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WADC TECHNICAL REPORT 55-337

FC

COMPARTMENTED FUEL TANKS

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O. W. GARD
W. W. WALTON

UNIVERSITY OF KENTUCKY

NOVEMBER 1953

WRIGHT AIR DEVELOPMENT CENTER

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POWER PLANT LABORATORY
CONTRACT No. AF 33(600)-22876
PROJECT No. 3084

WRIGHT AIR DEVELOPMENT CENTER
AIR RESEARCH AND DEVELOPMENT COMMAND
UNITED STATES AIR FORCE
WRIGHT-PATTERSON AIR FORCE BASE, OHIO

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FOREWORD

This report was prepared by Dr. W. M. Carter and Professor O. W. Gard of the Kentucky Research Foundation, University of Kentucky, Lexington, Kentucky, on Air Force Contract No. AF33(600)-22876, under Task No. 30272 "(U) Flexible Tanks." The work was administered under the direction of the Power Plant Laboratory, Directorate of Laboratories, Wright Air Development Center, with Lt. T. B. Cooper acting as Project Engineer.

The assistance of Dr. K. O. Lange, Director of the Aeronautical Research Laboratory; Professor E. B. Penrod, Head of the Mechanical Engineering Department; Messrs. W. W. Walton and M. K. Marshall of the Mechanical Engineering Department; and Messrs. E. R. Berry and A. R. Maddox, employees of the Aeronautical Research Laboratory in the preparation of this report is gratefully acknowledged.

ABSTRACT

A study on compartmented fuel tanks has been made to determine the optimum number of compartments which give maximum leakage protection for minimum weight.

Design criteria have been established to evaluate various arrangements, and several designs have been studied.

From the results of this analysis the two most promising styles were selected for further study. These are the cellular type and the vertical tube type. One of each has been designed and detailed.

It is concluded that compartmented fuel tanks can be built which give high leakage protection against 40 mm or larger missiles with a weight which compares favorably with the present self-sealing tanks.

PUBLICATION REVIEW

This report has been reviewed and is approved.

FOR THE COMMANDER:

E. C. Phillips
NORMAN C. APPOLD
Colonel, USAF
Chief, Power Plant Laboratory

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INTRODUCTION

One of the critical problems in military aviation is the storing of fuel in tanks of various sizes, located in the wings and other positions of the airframe. The vulnerability to enemy action of these tanks is very high, since they are usually of large size and are not protected by armor plate. As a result, any missile could severely cripple the plane if it penetrated the fuel tanks, by causing a loss of fuel. In order to protect the plane, the self-sealing gas tank was introduced. This is basically a synthetic rubber tank whose walls are made in layers with an inner layer of natural rubber. If a projectile pierces the tank, the escaping fuel causes the natural rubber to swell and seal the opening. This construction has proved to be satisfactory for missiles of 20 mm or smaller. However, for larger projectiles, the self-sealing tanks will not sufficiently seal, and other means must be found to protect the tanks from leakage. It has been proposed to build a tank with many compartments. Such a tank should have a much greater leakage protection than one with only one fuel space, since a penetrating missile could cause a fuel loss in only the compartments it damaged.

This report presents the results of a study that has been undertaken by the Kentucky Research Foundation, University of Kentucky, at the request of the Air Force (Contract AF 33(600)-22876) to determine the optimum arrangement of the cells and to design two tanks according to the results of the study.

SECTION I

DESIGN CRITERIA

In order to evaluate the efficiency of compartmented tanks it was necessary to establish definite means of determining the leakage under various conditions of damage. Since battle-damage conditions vary so widely, certain exact conditions that could be evaluated scientifically were set up as design criteria. In Exhibit B of the contract certain battle-damage conditions are listed. A summary of these conditions follow:

1. A 3/8" 75 ST aluminum plate shall be placed 2 in. from the tank. One round each of 20 mm HEl and 40 mm APl shall be fired through the aluminum plate at an angle of 45° to the normal. One round each of 20 mm APl and 40 mm HEl shall be fired through the aluminum plate at an angle of 30° to the normal. One round each of 20 mm HEl and 40 mm APl shall be fired through the aluminum plate at an angle of 0° to the normal.

2. A 1/8" 75ST aluminum plate shall be placed 2 in. from the tank. One round each of 20 mm HEl and 40 mm APl shall be fired through the aluminum plate at an angle of 80° to the normal. One round each of 20 mm APl and 40 mm HEl shall be fired through the aluminum plate at an angle of 45° to the normal. One round each of 20 mm HEl and 40 mm APl shall be fired through the aluminum plate at an angle of 0° to the normal.

3. Tests of aluminum plate and angle conditions as those of 1 and 2, but using two fragments of 1/4 oz. traveling at a velocity of 5000 ft/sec. shall be made in place of the 20 mm and 40 mm projectiles, for each of the angle positions.

4. Tests of aluminum plate and angle conditions as those of 1 and 2, but using a 2-1/2 in. diameter rod 6 in. long, traveling at a velocity of 2000 ft/sec. shall be made in place of the 20 mm and 40 mm projectiles, for each of the angle positions.

For each of these battle-conditions, five different capacities are to be used: 0, 20, 40, 80 and 100%. This study is to be conducted for structural weights which will withstand the following accelerated and static loads:

1. Vertical loads of 7.33g and -4g.
2. Side loads of 4g.
3. Fore and aft loads of 4g.

Each item in the study is to consist of a chart which shows the various weights of tanks vs the percentage leakage. (Percent leakage is defined as the ratio between the amount of fuel lost and the total amount in the tank before leakage, times 100).

In order to plot these curves, it is necessary to devise a certain structure and calculate its weight. Then the percent leakage can be determind for the specific conditions, and one point on the curve can be found. It is then necessary to change the number of divisions in the tank and recalculate its weight. The new percent leakage for this conditions is determined and plotted, thus forming the chart for the particular tank style selected.

After considerable thought and study on the problem of determining the charts for each of the conditions outlined above, a technical conference was held between the contractor and WADC to clarify the matter and agree upon a practical program of determining the percent leakage curves. The following points were established as a working program for this study:

1. The evaluation of a tank for leakage protection will not necessarily follow the same pattern as the testing procedure after the tank has been constructed. Since it is desirable to rate the various designs on the same basis, a clarification of the shot patterns suggested in Exhibit B of the contract is needed. This exhibit specifies the angle to the normal only, at which the projectiles strike the plate placed in front of the tank. Hence, to control the shot pattern more closely, the direction and angles shown in Fig. 1 were established for location, direction and number of shots. These angles all agree with the standards set up in Exhibit B, but impose an additional specification condering the location of the shots.

2. There is little information available concerning the distance various missiles will travel through gasoline after penetrating the aluminum plate placed in front of the tanks. It has been observed that it takes approximately 30 in. of water to stop a 50 caliber projectile. Thus it seems reasonable to assume that the projectiles to be used in this study would travel at least the same distance through gasoline. However, the HE ammunition, as well as the fragments, have an unpredictable effect. Since a standard for evaluating the tanks is desired, exact penetration will not be important if it is the same for all missiles used.

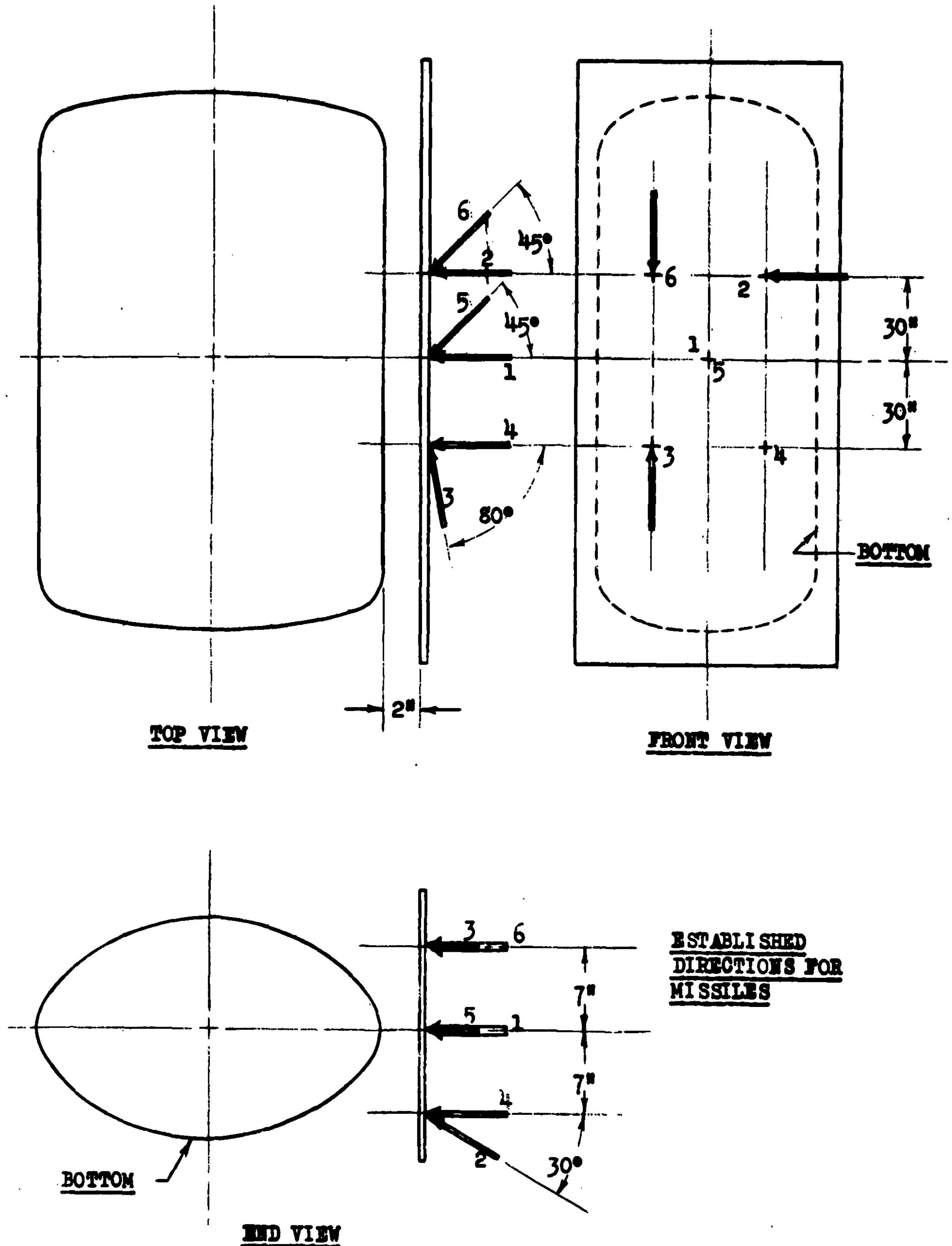


FIG. 1 STANDARD SHOT PATTERN

Hence, it was assumed that the projectiles would go completely through the tanks.

3. It was further agreed that the tanks should be oval in shape, contain 640 gallons and approximate the dimensions of the tank shown in U.S. Air Force Drawing No. S52J9341.

After establishing this program, scale drawings were made of the tank with the paths of the projectiles clearly shown. By dividing the space of the tank into compartments, it was possible to determine from the three views of the tank how many compartments would be damaged by the projectiles. Then the percentage leakage could be calculated. For the different style tanks the weights could be estimated, from which it was possible to draw the percent leakage vs weight curves. This was done for all the types investigated, and then the styles which gave the greatest protection for the smallest weight were further examined. In making this study the following principles were formulated as a guide for tank design:

1. The largest number of individual compartments gives the greatest protection, regardless of how those compartments are shaped or placed.

2. The space between the compartments must be held to a minimum to insure leakage protection.

3. Each compartment must be individually connected to a sump or provided with check valves, to prevent back flow from other compartments in the event of damage.

4. Compartments should be arranged to enable the collection of any leakage into a separate lower compartment, if possible.

5. The sump should be as small as possible to decrease the chance of damage from a missile.

The use of such a standard procedure greatly clarifies the problem of making the design study, since it permits the percent leakage to be obtained with some degree of accuracy. It is to be expected that the tanks will be at least as safe as theoretically indicated since it is expected that the design criteria impose a greater damage condition than would actually result.

SECTION II

EVALUATION

CELLULAR TANK:

There are many methods by which a cellular tank may be constructed. It becomes necessary to determine some one method which is practical and at the same time fulfills all the requirements set forth in the previous section on design criteria. There are several shapes which can be used for the cells. To satisfy the requirement that the volume between the compartments be held to a minimum, it seems desirable to select a shape which would "stack" in space; that is, one in which there would not be any large volume between the cells. For conservation of material it also becomes evident that a shape which has the largest ratio of volume to surface would result in the lightest design. This would indicate a shape that approaches a sphere, but at the same time completely fills the volume of the tank.

The only regular polyhedrons which will do this are the dodecahedron, composed of twelve pentagons, and the cube, composed of six squares. Although the dodecahedron appears very attractive from a volume-surface ratio consideration, the attending difficulties of construction and plumbing make the cube the logical choice. Hence the cellular tank, which represents the construction with the largest number of compartments, was designed using the cube as the basic building unit.

The next step was to design a reasonable structure in which the cubes could be supported and to devise some method by which the fuel could be delivered to the sump from each cell individually. Although the following design has been altered somewhat in the final layouts, it was used as a basis for calculating the tank weights. This was necessary as a start, since the weight of the tank was required before the percent leakage versus tank weight charts could be constructed.

The basic unit of this design is a cell of approximately cubical shape, open at the top, and constructed of bladder material. Fig. 2 shows both the developed and folded views. Each cell holds approximately one-half gallon for the edge dimension chosen, there being about 1200 of these cells in the whole tank. This corresponds to an edge dimension of approximately 5 in. which was arbitrarily chosen as a starting point for the design.

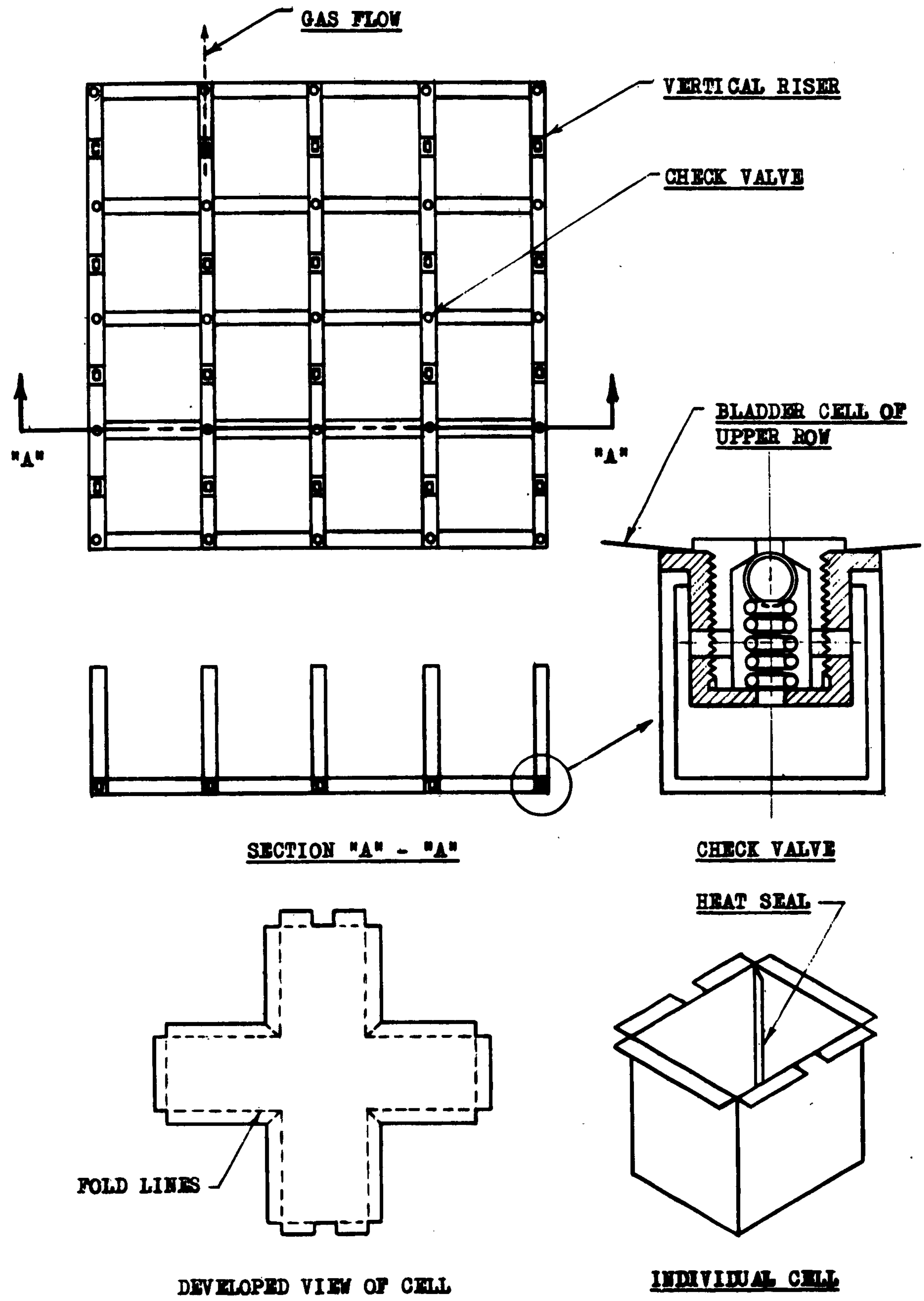


FIG. 2 FRAME AND CELL CONSTRUCTION

These cells are supported on a framework of 3/8 in. square aluminum tubing, with 0.022 in. wall thickness. Fig. 2 shows a drawing of one bank of this tubing structure. The drains from the individual cells are formed by the structure itself and connect to a header in the center of the tank. There is a header for each layer. These connect in turn to a sump in the bottom of the tank, where a booster pump is located. A float valve on the bottom of each line from the cells permits the layer so connected to discharge into the sump independently of the other layers.

To assemble this structure, the lower tubing bank is placed in position. Then the next layer is placed on top and riveted down. The cells are then inserted. The upper lip of each cell forms a support for the whole cell, being attached to the next cell by a heat seal. Small unit check valves are then inserted in the bottom of the cell, from the top, and serve to fasten the cell to the lower frame, while at the same time providing a passage way into the lower structure. This lower structure is then connected by a header to a line which runs in the center of the tank to the sump.

The next layer is then added and the process described above is repeated until the whole tank is built up. The sump is placed in position and the connecting lines from the header installed. Finally, the whole frame is wrapped in bladder material. This design meets most of the requirements listed under Section I, namely:

1. Large number of cells.
2. Small space outside cells.
3. Each cell individually connected to a sump.
4. Compartments arranged so that lower ones catch fuel lost from upper.
5. Small sump

A very essential quantity to be determined is the weight of the tanks for various cell sizes. This weight must be known before the charts of Percent Leakage vs Weight can be drawn. An estimate for the weights may be made as follows:

Let the capacity of the tank = 640 gallons = 148,000 cu. in.

Then $W = w_1 + w_2 + w_3 + 50$

Where W = Total tank weight-lbs.

w_1 = weight of cells-lbs.

w_2 = weight of tubing-lbs.

w_3 = weight of check valves-lbs.

50 = assumed weight of sump plus pump plus covering-lbs.

These values may be calculated from the following equations:

$$w_1 = NS(.06)/(144)$$
$$w_2 = 3\ a\ Nw'$$
$$w_3 = N(.03)$$

Where

N = number of cells
S = surface of cell-sq.in.
$.06$ = weight of bladder material-lbs/sq.ft.
144 = conversion factor
a = length of a cell side-in.
w' = weight of unit length of tubing-lb/in
T = side dimension of square tubing
$.03$ = unit weight of check valve-lbs.
$S = 5(a-T)^2$
$N = (640)(231)/a^3$

These weights have been estimated from the preliminary design. The weight of the final tank was higher, mainly because of a difference in construction and material choice. However, weights as calculated from the above relationships will serve adequately as a basis for drawing the Percent Leakage vs Weight curves for this particular style tank. Having then set up means for determining the tank weight for a given cell size, the following table was constructed to extend these calculations to cell sizes ranging from 5 to 20 in.:

TABLE I

WEIGHT SUMMARY OF CELLULAR TANKS

a	N	a^3	$(a-T)$	$(a-T)^2$	S	w_1	T	w'	w_2	w_3	W
5	1180	125	4.62	21.34	106	39.3	3/8	.00308	54.5	35.4	179
6	680	216	5.62	31.58	157	35.0	3/8	.00308	37.7	20.4	143
7	430	343	6.62	43.82	219	32.2	3/8	.00308	28.4	12.9	123
8	290	512	7.50	56.25	281	29.5	1/2	.00500	34.8	8.7	123
9	200	729	8.50	72.25	361	26.7	1/2	.00500	27.0	6.0	109
10	148	1000	9.50	90.25	451	25.0	1/2	.00500	22.2	4.5	101
11	110	1331	10.34	106.91	534	22.9	5/8	.00558	20.3	3.3	96
12	86	1728	11.34	128.59	642	21.7	5/8	.00558	17.3	2.6	91
13	68	2197	12.34	152.27	761	20.4	5/8	.00558	14.8	2.0	87
14	54	2744	13.25	175.56	877	19.0	3/4	.00958	21.7	1.6	92
15	44	3375	14.25	203.06	1015	17.9	3/4	.00958	19.0	1.3	88
16	36	4096	15.25	232.56	1162	16.9	3/4	.00958	16.5	1.1	84
17	30	4913	16.00	256.00	1280	16.0	1	.0223	34.1	.9	101
[illegible]	25	5832	17.00	289.00	1445	15.0	1	.0223	30.1	.7	95
19	21	6859	18.00	324.00	1620	14.1	1	.0223	26.7	.6	91
20	18	8000	19.00	361.00	1805	13.5	1	.0223	24.1	.5	88

Figs. 3 and 4 show the data of Table I in graphical form.

To complete the Percent Leakage vs Weight curves it was necessary to calculate the percent leakage. This was accomplished by making scale drawings of the tanks and superimposing the standard shot pattern of Fig. 1. By inspection, the number of damaged cells was determined. Fig. 5 is a sample drawing showing the shot pattern as applied to a tank with a 5 in. cube. To determine the percent leakage, two conditions must be considered: first, when the tank is full; and second, when the tank is less than full.

For the case where the tank is full:

Let
P = percent leakage
V = intercellular volume-gallons
G = cell volume-gallons
X = number of cells damaged

Then
$$P = \frac{XG + 0.75V}{640} \qquad (1)$$

The intercellular volume V = (volume in tank) -(volume in cells)

$$V = \left[148,000 - Na(a-T)^2\right] /231$$

For the case where the tank is less than full:

Let C = percent tank capacity

Then
$$P = \frac{\left[1 - \frac{(640)(1-C)-V}{XG}\right] XG - \frac{V}{4}}{640C}$$

(Due to the arrangement of the cells, any fuel lost from a damaged cell is collected in the bottom of the tank, up to 1/4 the intercellular volume.)

But
$$NG + V - 640 = 0$$

$$P = \frac{\frac{640CX}{N} - \frac{V}{4}}{640C}$$

and
$$P = \frac{X}{N} - \frac{V}{2560C} \qquad (2)$$

Fig. 6 is an alignment chart for equation (1) and Fig. 7 is an alignment chart for equation (2). By the use of these alignment charts, the percent leakage could be calculated when the number of damaged cells were known for various percent capacities.

Table 2 is a summary of the calculations carried out as above for determining the percent leakage for the various arrangements of the cellular tanks.

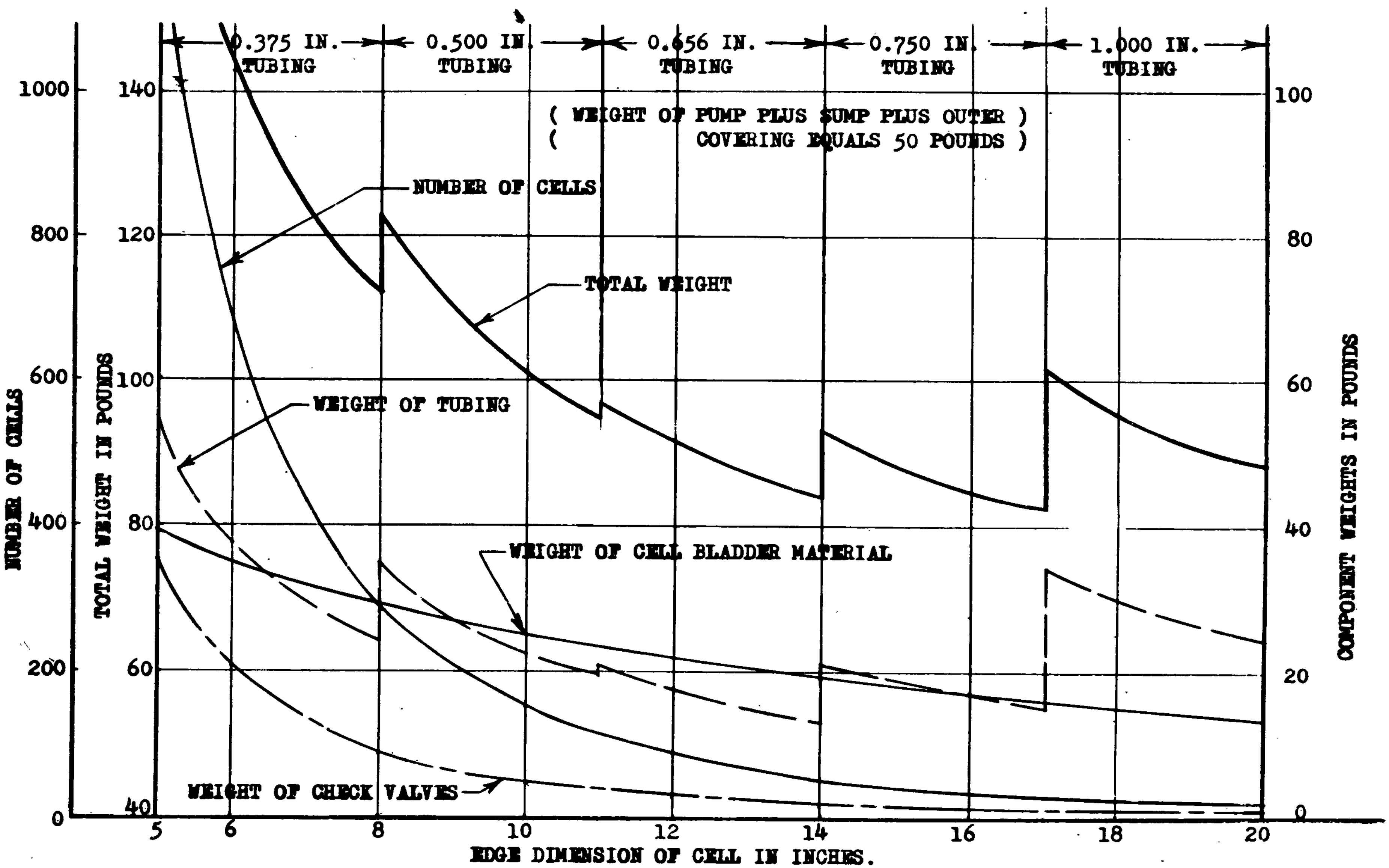


FIG. 3 . WEIGHTS AND NUMBER OF CELLS VERSUS EDGE DIMENSION OF CELL.

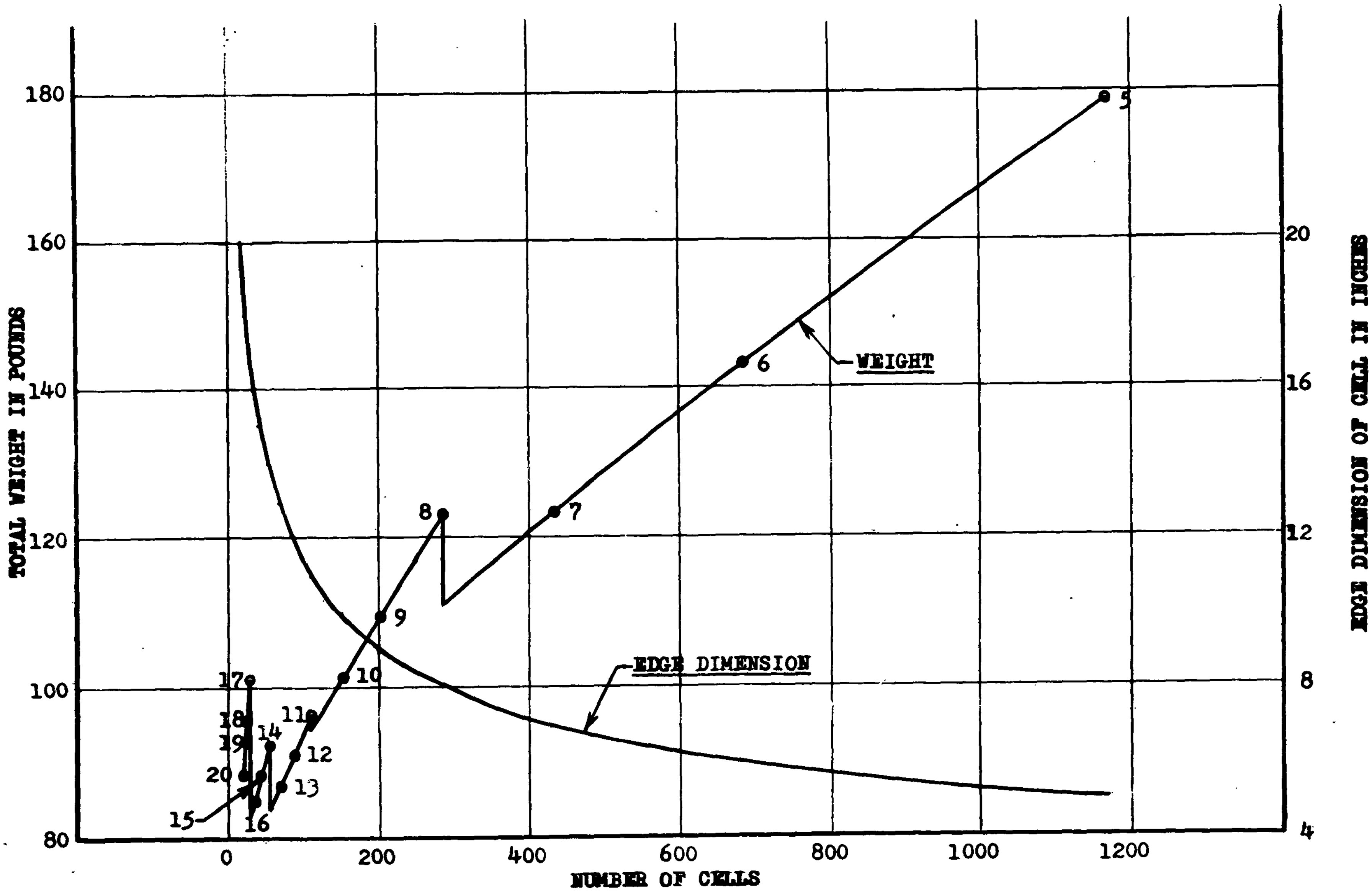


FIG. 4. WEIGHT AND EDGE DIMENSION VERSUS NUMBER OF CELLS.

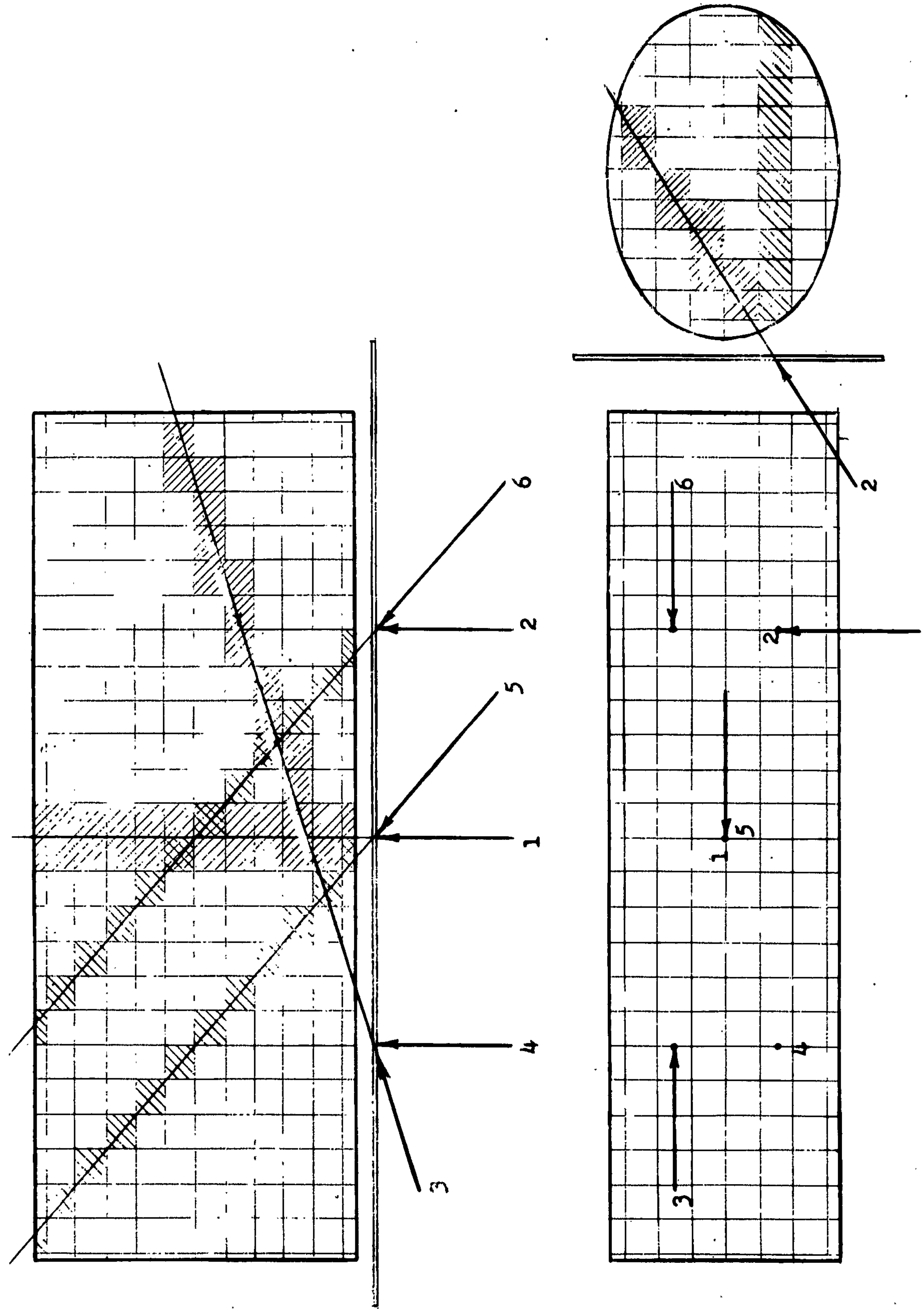


FIG. 5 SAMPLE LEAKAGE EVALUATION

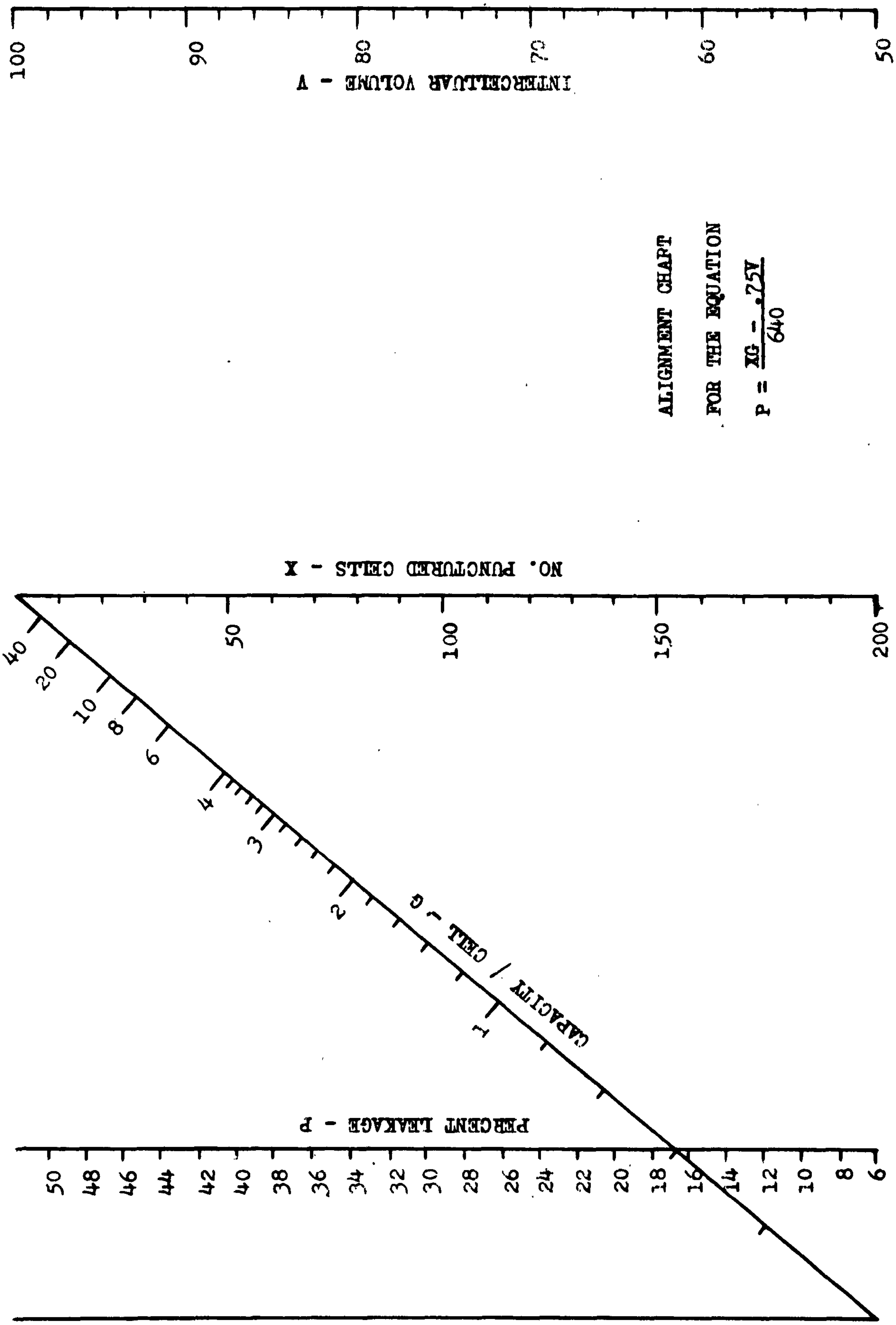


FIG. 6 ALIGNMENT CHART NO. 1

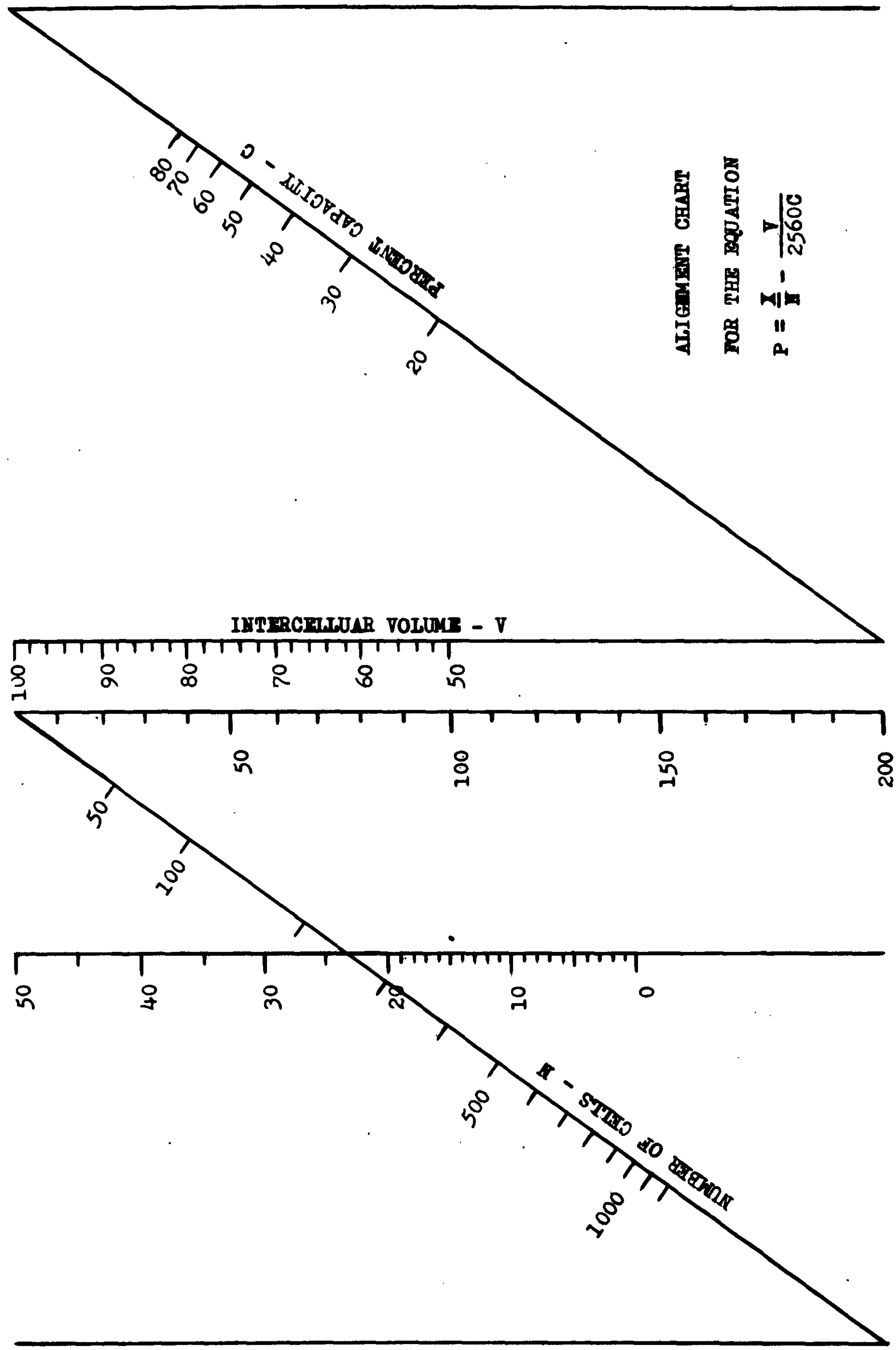


FIG.7 ALIGNMENT CHART NO. 2

TABLE 2

SUMMARY-CELLULAR TYPE

a Cell Edge	N No. Cells	W Weight	G Gal. Cell	X No. Punct. Cells	V Extrn. Gal.	C % Cap.	P % Leak.
				156		100	22.4
5	1180	179.2	.462	110	95	80	5.0
				97		60	2.0
				96		40	0
				86		20	0
				130		100	26.4
6	680	143.1	.82	85		80	9.0
				75		82.5	5.6
				69		40	1.8
				67		20	0
				144		100	37.5
				94		80	18.6
7	430	123.5	1.327	76	70.5	60	12.5
				72		40	10.5
				72		20	3.0
				79		100	33.1
				55		80	16
8	290	123.0	1.945	54	75.6	60	14.5
				35		40	5.2
				33		20	0
				66		100	38.5
				43		80	18
9	200	109.7	2.810	31	77.8	60	11
				26		40	5
				24		20	0
				74		100	54
				55		80	50
				53		60	42
10	148	101.7	3.905	29	64.8	40	14
				27		20	7.5
				45		100	44.5
				30		80	36
11	110	96.5	5.08	29	81.2	60	22
				18		40	8
				17		20	0
				45		100	55
				30		80	32
12	86	91.6	6.60	29	67.0	60	30
				17		40	15
				17		20	9

Values of cube edges greater than 12 in. were not considered since the percent leakages became very high. It was felt that such a tank would not be too practical. Fig. 8 shows a graphical plot of the data in Table 2. This is the Percent Leakage vs Weight curves for the Cellular Type Tank.

It may be seen from Fig. 8 that the most advantageous cell size to use lies somewhere between 6 and 8 inches. On this portion of the curve the greatest gain in leakage protection is obtained for a given increase in weight with a reasonable value of percent leakage. It should be pointed out that the weights used in plotting these curves are approximate. However, since the same methods were used for obtaining the weights of all tanks, the curves give the true picture for establishing the optimum cell size.

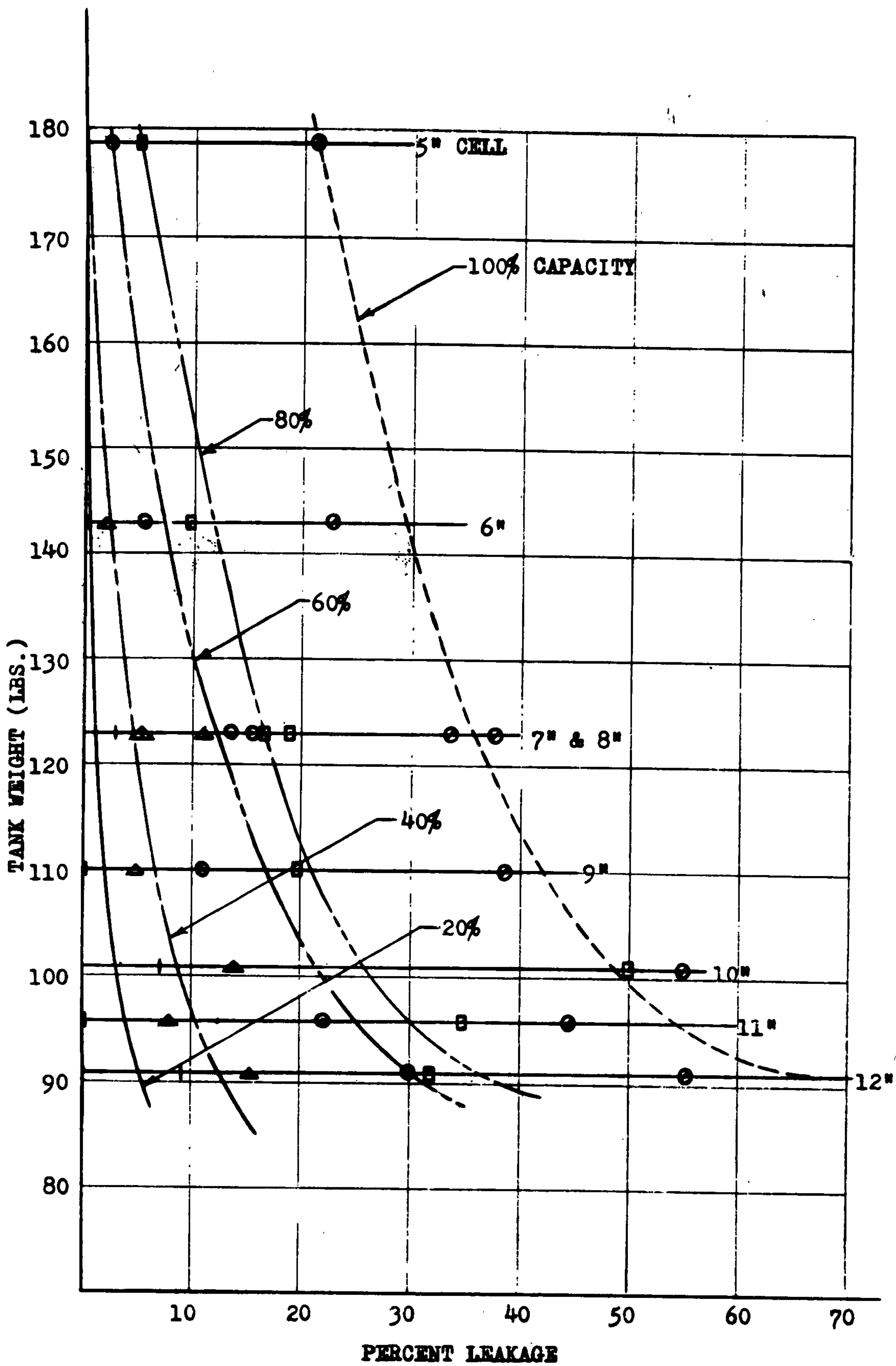


FIG. 8 % LEAKAGE VS. TANK WEIGHT FOR CELLUAR TYPE

HORIZONTAL TUBE TANK:

The first design of a horizontal tube tank was one with thin aluminum tubes supported at the ends by punched aluminum support plates and at intervals by bladder material separators. (See Fig. 9). Each tube was fitted with a check valve so that it acted as an independent compartment. Tube sizes considered were 2, 3, 4, 5, 6 and 8 inch outside diameters. A cursory examination revealed that the best leakage protection resulted by having a maximum number of independent compartments, thereby decreasing the amount of unprotected fuel in the external volume. Using this as one criteria, layouts were made in order to determine the number of tubes. The maximum number of tubes that could be placed in the given tank cross-section and still leave sufficient bladder material for support are shown in Table 5. A filler space of approximately six inches length and a sump of approximately ten inches length were located on opposite ends of the tank. This resulted in a tube length of 106 inches. A commercial size tube was selected to obtain the smallest wall thickness for the particular tube size in question. Table 3 shows the pertinent data for the sizes selected.

TABLE 3

ALUMINUM TUBE DATA

O. D.	Stubs Gage	Thickness in inches	Weight per inch	Ratio of gallons in tube to wgt. of tube
2	26	0.018	0.0113	1.16
3	24	0.022	0.0208	1.43
4	22	0.028	0.0350	1.51
5	20	0.035	0.0550	1.50
6	19	0.042	0.0785	1.52
8	-	0.046	0.1150	1.85

The span or distance between each bladder separator was calculated by considering the tube as a uniformly loaded simple beam and limiting the deflection (sag) to 0.015 inch. a sample calculation using the 8-inch O.D. tube is shown below:

$$l = \sqrt[4]{\frac{384\ dEI}{5v\ \ mn}} = \sqrt[4]{\frac{384 \times 0.015 \times 10^7 \times 9.06}{5 \times 0.2125 \times 6 \times 7.33}} = 58''$$

$$S_t = \frac{vmnl^2c}{8I} = \frac{0.2125 \times 6 \times 7.33 \times (56)^2 \times 4}{8 \times 9.06} = 1730 \text{ psi}$$

$$S_s = \frac{vmnl}{2A} = \frac{0.2125 \times 6 \times 7.33 \times 56}{2 \times 1.15} = 235 \text{ psi}$$

Where
- l = span length in inches
- d = deflection in inches
- E = modulus of elasticity in psi
- I = moment of inertia of tube about centroid in in^4

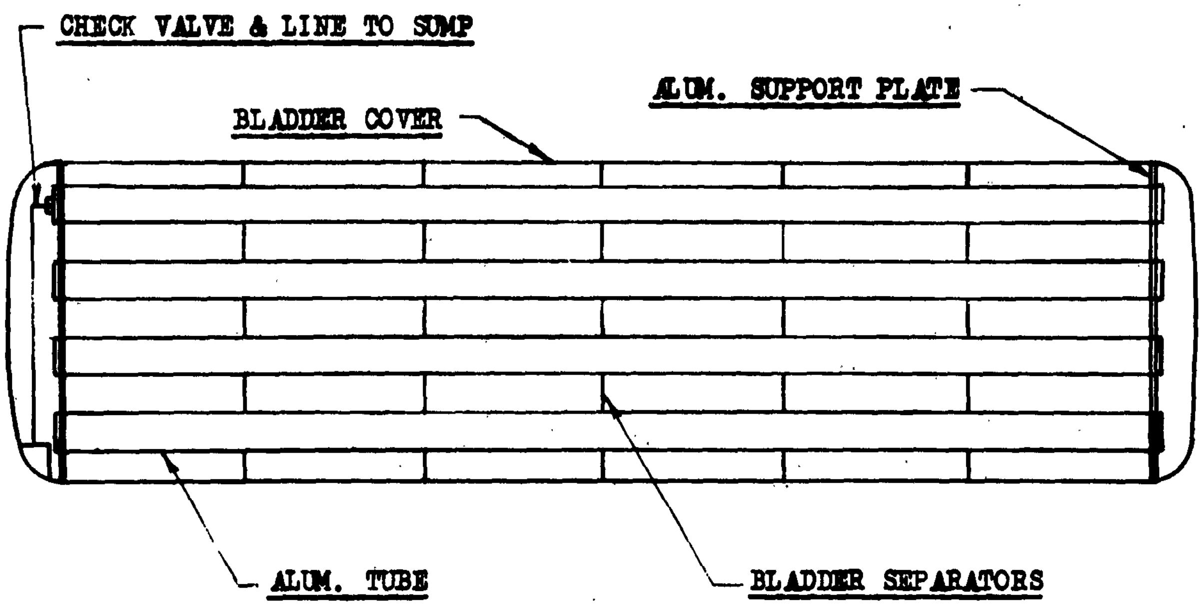


FIG. 9 HORIZONTAL ALUMINUM TUBE TANK-DESIGN ONE

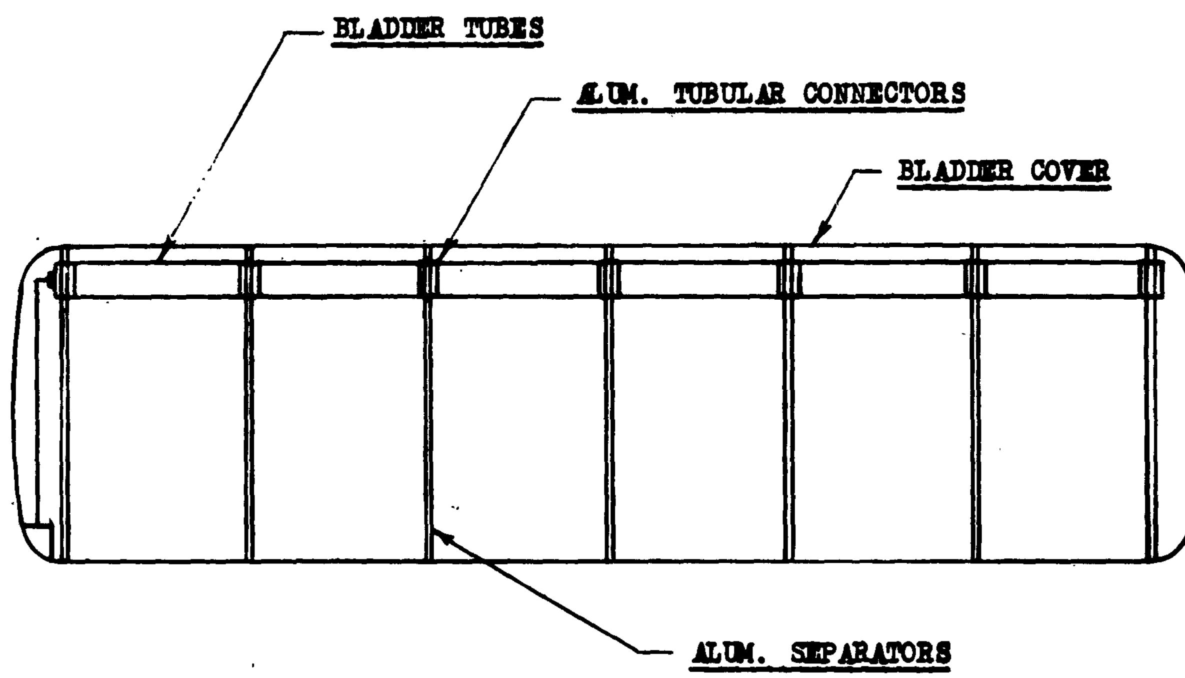


FIG. 10 HORIZONTAL BLADDER TUBE TANK, DESIGN TWO

V = gallons per inch of tube
m = weight of Fuel in pounds per gallon
n = ratio of acceleration to acceleration due to gravity
S_t = maximum unit tensile stress in psi
c = distance from centroid of tube to outer tube fiber in inches
S_s = maximum unit shear stress in psi
A = area of metal in cross-section of tube in inches^2

For the 2" O.D. aluminum tube the limiting span was 32", S_t was 1345 psi, and S_s was 82 psi. The weight of all tubes was calculated and found to vary from 297 pounds for 2" OD tube to 195 pounds for the 8" O.D. tubes. A bladder material sheet was selected for the shell of the tank which amounted to 8 pounds. The total weight of the internal supports was 15 pounds for the end aluminum supports and 4 pounds for the bladder separators. Ten horizontal compression tubes were used to tie together the two aluminum supports and to give some longitudinal strength. The weight of these compression tubes was 31 pounds. The weight of the lines and check valves were estimated at 0.10 pounds per tube. Allowing a weight of 20 pounds for the pump, float valve, filler caps, and vent and drain connections gave a weight of the complete tank as shown in Table 5, page 26.

The Percent Leakage was found by drawing the cross-sectioned end and plan views of the tank showing the tubes and the bladder separators which formed the walls of the external compartment.

From these views, the number of tubes and compartments affected by each shot could be observed. Since each tube ran the length of the tank it was considered to be emptied by one hit and further hits had no effect. It was also necessary to take into account the amount of fuel from the damaged tube which could be retained by the external compartments. The following shows the method of calculating the Percent Leakage for the 4" O.D. Tube.

1. <u>Tank Full (640 gallons)</u>

Volume per tube 6.08 gallons
Number of tubes = 56
Volume in tubes = 340 gallons
Volume in external compartment = 300 gallons
No. of external compartments = 7
Volume per external compartment = 43 gallons

Shot Number	Full Tubes hit	Percent Fuel in Compartment Lost	Total Fuel Lost
1	12	0.50	94.4
2	4	0.75	65.4
3	4	4 x 0.25	67.1
4	8	0.50	70.1
5	0	2 x 0.5	42.8
6	3	0	18.2
			349.0

% Leakage = $\frac{346}{640}$ x 100 = 54.5%

2. Tank 80% Full (512 gallons)

Volume in tubes = 340 gallons
Volume in external compartment = 172 gallons
Volume per external compartment = 24.6 gallons

Shot Number	Full Tubes Hit	Percent Fuel in Compartment Lost	Total Fuel Lost
1	12	0	76.0
2	4	0.25	35.0
3	4-2*	0	13.2
4	8	0.25	59.4
5	0	0.25	13.7
6	3	0	18.2
			215.5

% Leakage = $\frac{215.5}{512}$ x 100 = 42.1%

* Indicates that the fuel from two tubes was caught and retained in the external compartment.

3. Tank 60% Full (384 gallons)

Volume in tubes = 340 gallons
Volume in external compartment = 44 gallons
Volume per external compartment = 6.3 gallons

Shot Number	Full Tubes Hit	Percent Fuel in Compartment Lost	Total Fuel Lost
1	12-2	0	60.8
2	4-1	0	18.2
3	4-3	0	6.1
4	8-1	0	42.6
5	0	0.25	10.7
6	3	0	18.2
			156.6

% Leakage = $\frac{156.6}{384}$ x 100 = 40.5

4. Tank 40% Full (256 gallons)

Volume in tubes = 256 gallons
Number of tubes filled = 42
Volume in external compartment = 0

Shot Number	Full Tubes Hit	Percent Fuel in Compartment Lost	Total Fuel Lost
1	12-3.6	0	51.0
2	2-2	0	0
3	0	0	0
4	8-1.8	0	37.8
5	0	0	0
6	0	0	0
			88.8

$\% \text{ Leakage} = \frac{88.8}{256} \times 100 = 34.7$

5. Tank 20% Full (128 gallons)

Volume in tubes = 128 gallons
Number of tubes filled = 21
Volume in external compartment = 0

Shot Number	Full Tubes Hit	Percent Fuel in Compartment Lost	Total Fuel Lost
1	0	0	0
2	1 - 1	0	0
3	0	0	0
4	8 - 1.8	0	37.8
5	0	0	0
6	0	0	0
			37.8

$\% \text{ Leakage} = \frac{37.8}{128} \times 100 = 29.5$

The following Table shows the Percent Leakage at various Percent Capacities of the tank for the first design of horizontal tube tanks.

TABLE 4

PERCENT LEAKAGE FOR HORIZONTAL ALUMINUM TUBE TANK

Tube O.D. inches	2	3	4	5	6	8
Percent Capacity						
100%	37.3	40.7	54.5	61.8	72.8	80.3
80	23.6	35.5	42.1	54.0	63.0	79.3
60	18.0	34.3	40.5	55.1	59.1	90.3
40	16.6	32.6	34.7	46.2	69.3	89.0
20	15.2	31.3	29.5	30.8	69.8	93.3

Since the weight using aluminum tubes was excessive, an effort was made to use a lighter tube material since a large proportion of the weight was contributed by the tubes.

In this second design a bladder material was used in place of the thin aluminum tubes (See Fig. 10.). Because sagging of the tube would result in a large quantity of fuel that could not be drained from the tubes, it was necessary to have six or more connectors which were supported and held in fixed position by rigid punched aluminum separators. The weight of these tubes, connectors, and aluminum separator was less than the thin wall aluminum tube for all tube sizes. Since the tube spacing and arrangement was the same as was used in the aluminum design, the Percent Leakage would be the same as is shown in Table 4.

Figure 11 shows the Percent Leakage versus Tank Weight. Heavier structural members for the 6" O.D. Tube Tank would have resulted in a marked irregularity of the curves. Therefore the curves were not extending through the points representing the 6" O.D. Tubes.

The weights of Tanks for the Bladder Tubing Design are shown in Table 5.

In this design the connectors consisted of short tubes 2 in. long on which the flexible bladder tubes were clamped by a metallic strip. The connectors in turn were spot welded to the aluminum separators which were 10 gage (B&S) aluminum sheet. The remainder of the tank was the same as the aluminum tube design.

The bladder tube design showed a much lower weight than the original horizontal tube design, but there was introduced a significant problem of assembly and maintenance of non-sagging bladder tubes. As an approach to this problem, a rigid support for the bladder tubes in the form of aluminum half tubes was used in this third design. The span or distance between supports was calculated in the same manner as was done for the thin wall aluminum tubes in the first design. Since the Percent Leakage for the preceding designs was high, the number of tubes with separate connections was doubled by having the tubes in two sections, each section running half the length of the tank (See Fig. 12).

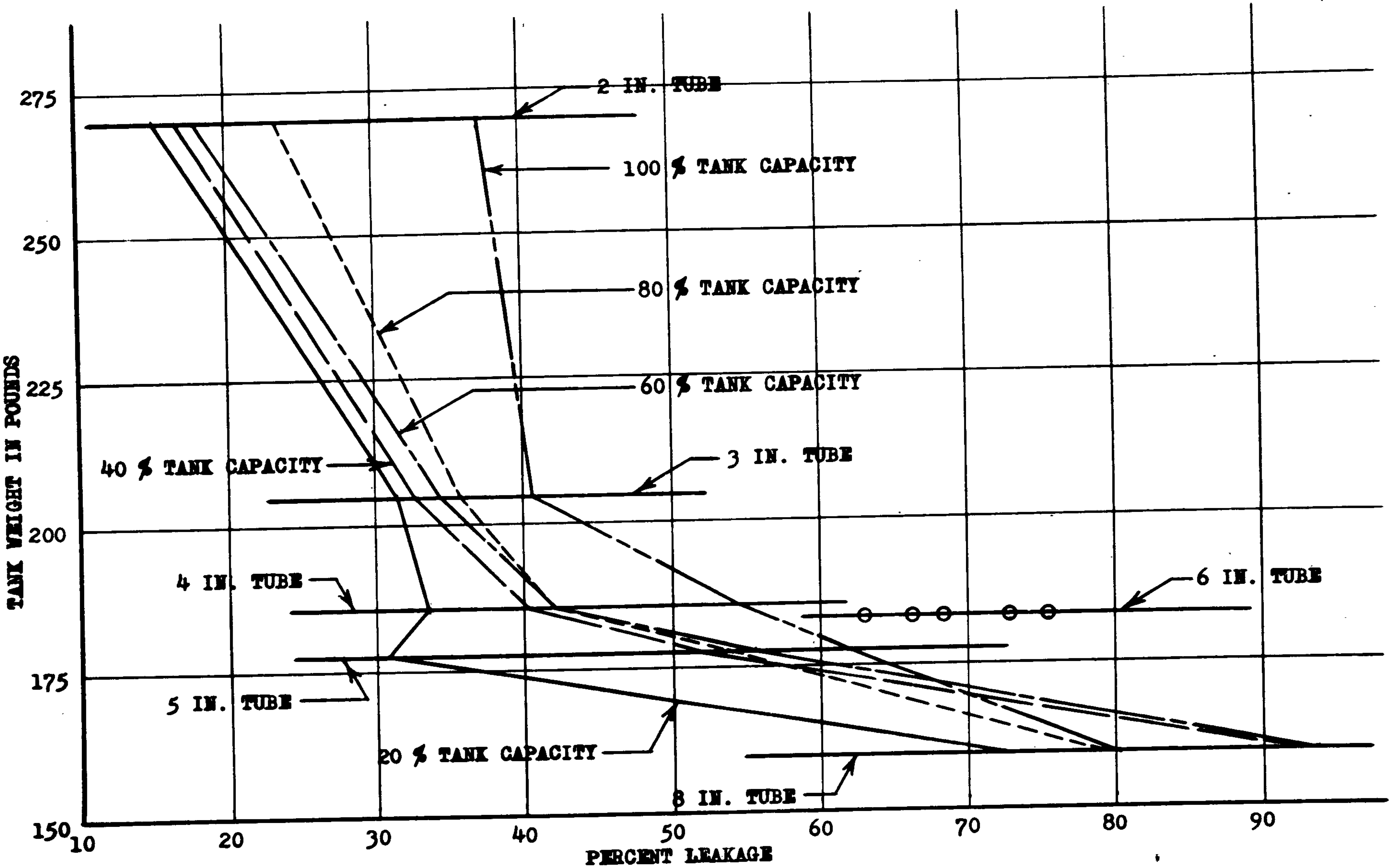


FIG. 11. PERCENT LEAKAGE VERSUS TANK WEIGHT FOR BLADDER TUBE.

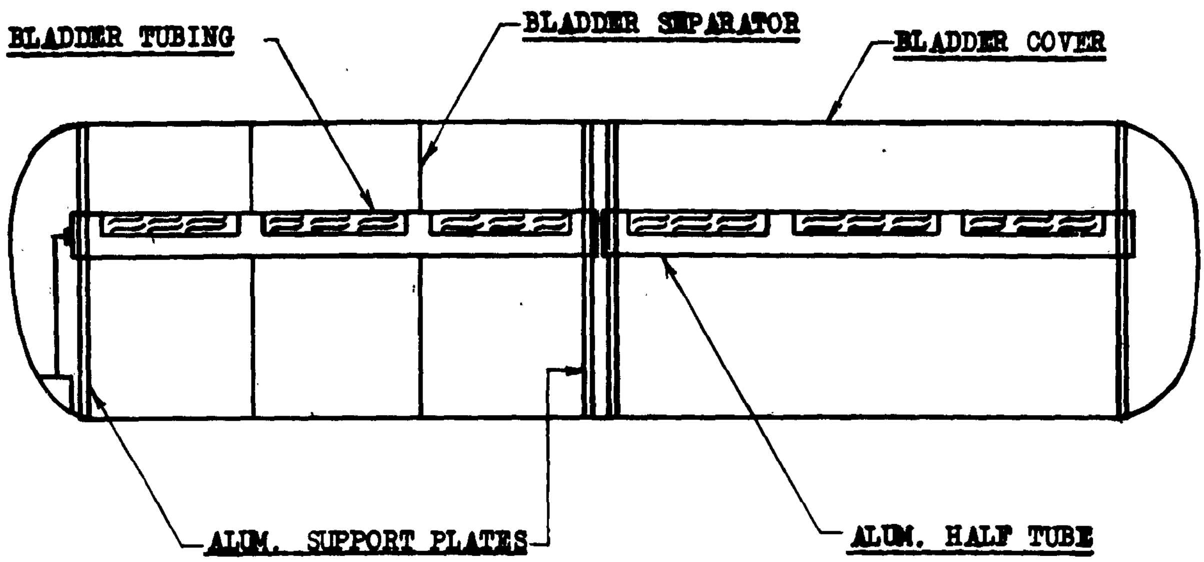


FIG. 12 HORIZONTAL BLADDER TUBE TANK, DESIGN THREE

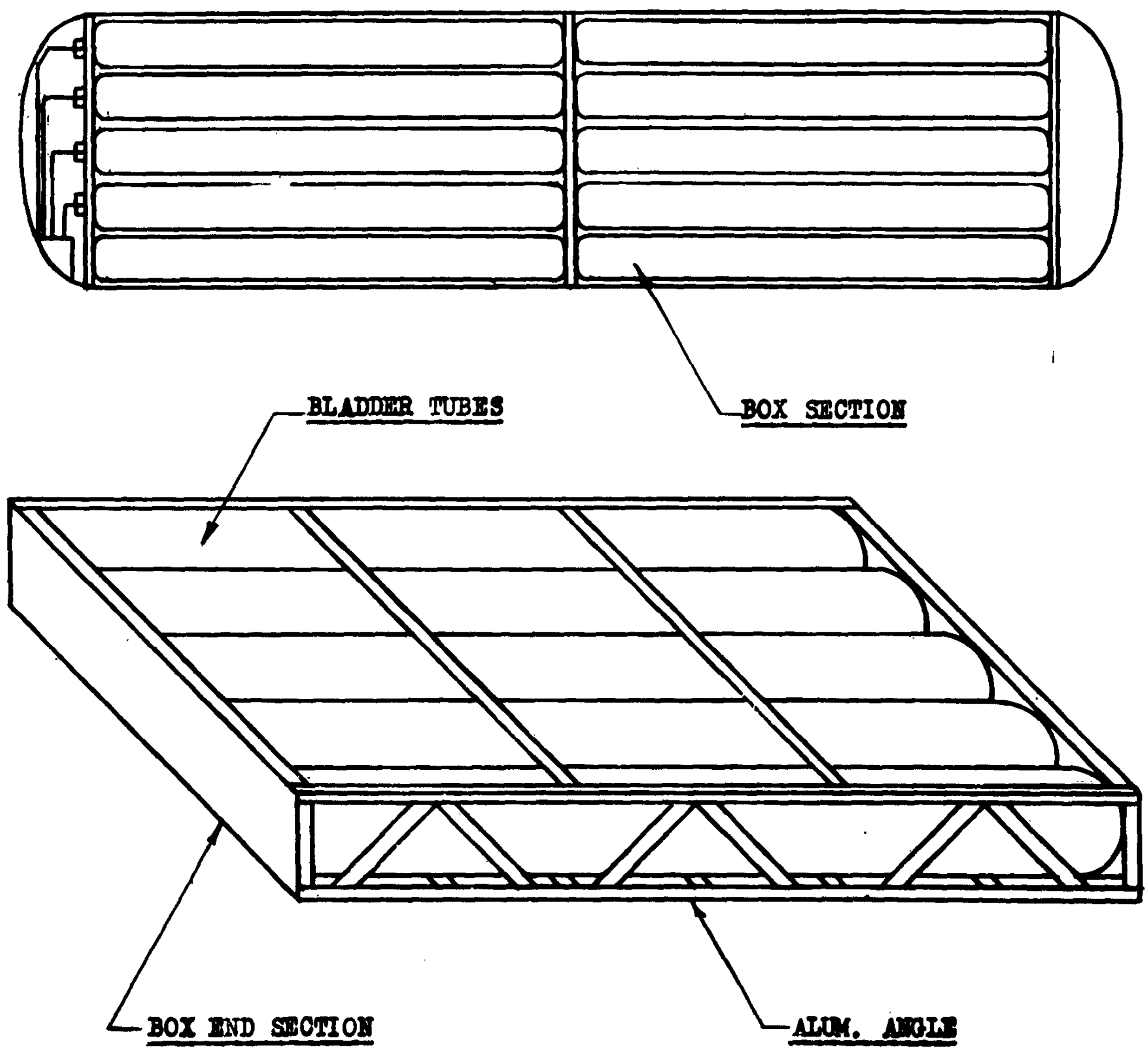


FIG. 13 HORIZONTAL BLADDER TUBE TANK, DESIGN FOUR

TABLE 5

DATA FOR HORIZONTAL TUBE TANKS

	Tube Size O.D. (inches)	Aluminum Tubing	Bladder Tubing With Connectors	Bladder Tubing With Half Tube	Bladder Tube Box
Weight of Tank (Pounds)	2	400	269	338	358
	3	342	204	275	285
	4	292	185	225	240
	5	303	177	226	211
	6	398	183	223	205
	8	275	159	194	182
Number of Tubes (with separate connection to sump)	2	248	248	496	372
	3	114	114	228	188
	4	56	56	112	104
	5	38	38	76	66
	6	26	26	52	48
	8	16	16	32	24
Capacity in tubes (gallons)	2	373	373	373	283
	3	390	390	390	324
	4	340	340	340	316
	5	360	360	360	314
	6	352	352	352	329
	8	389	389	389	292

The weight of the connectors used in the second design (bladder tubes with connectors) was eliminated but the half tubes introduced a significant increase in weight as is shown in Table 5. Although the protection would obviously be improved and the problem of sagging tubes largely eliminated, it was felt that the problem of assembly was further complicated. This third design (bladder tubes with aluminum half tubes for support) was eliminated in favor of one having greater ease of assembly and maintenance.

The fourth design of horizontal tubes consisted of tubes made of bladder material around which was constructed a box or framework of light aluminum angles and formed aluminum sheet (See Fig. 13). These box sections which were 56 inches in length were independent units which could be assembled with relative ease. Each tube within the box section contained its own check valve so as to give a maximum number of independent units. Stability was provided by vertical rods which tied together the box section. These vertical rods in turn were fastened to the tank structural member which were the same as was used in the three previous designs. The widths and number of box sections was obtained

from a layout drawing of the tank cross section for each size tube considered. Two different size box sections were used for each tube size in order to utilize as much of the available space as possible. Having determined the number of tubes in each box section, the sizes of angles were determined by considering the box structure as a fixed-end beam uniformly loaded. The following shows the calculation of the stress in the aluminum angles for the 4 inch O.D. Tube sections:

<u>Six Tubes per Box Section</u> (1 x 1 x 1/16 angles assumed)

$$S_t = \frac{Nvmnl^2c}{12\ I} = \frac{6 \times 0.054 \times 6 \times 7.33 \times 56^2 \times 2.56}{12 \times 0.652}$$

$$= 14{,}700 \text{ psi}$$

$$d = \frac{Nvmnl^4}{384\ EI} = \frac{6 \times 0.054 \times 6 \times 7.33 \times 56^4}{384 \times 10^7 \times 0.652} = 0.055 \text{ inches}$$

<u>Four Tubes per Box Section</u> (3/4 x 3/4 x 1/16 angles assumed)

$$S_t = \frac{4 \times 0.054 \times 6 \times 7.33 \times 56^2 \times 2.56}{12 \times 0.50} = 12{,}300 \text{ psi}$$

$$d = \frac{4 \times 0.054 \times 6 \times 7.33 \times 56^4}{384 \times 10^7 \times 0.50} = 0.046 \text{ inches}$$

S_t = maximum unit tensile stress in psi
N = Number of tubes in box section
v = gallons per inch of tube
m = weight of fuel in pounds per gallon
n = ratio of acceleration to acceleration due to gravity
l = span length in inches
c = distance from centroid of box to outer fiber of angle in inches
I = moment of inertia of four angles about centroid of box section
E = modulus of elasticity in psi

Sagging of the bladder tubes was prevented by having corrugated strips across the bottom of each box section. The number and size of these strips was calculated in the same manner as size of the angles in the preceding calculations. The box section ends were formed from 20 gage (B & S) aluminum sheet and designed to interlock with the box sections below and above.

Figure 5 shows the weights of the bladder tube box tanks. It will be noted that these weights are higher than those for

the bladder tubing with connectors design and that the amount of fuel in the tubes is less.

The Percent Leakage for the bladder tubing box design, which was calculated in the same manner as for the first design of horizontal tubes, is shown below and in Fig. 14.

TABLE 6

PERCENT LEAKAGE FOR BLADDER TUBE-DESIGN FOUR

Tube O.D. (inches)	2	3	4	5	6	8
Percent Leakage %						
100% full	37.9	39.2	43.0	54.3	48.8	64.8
80% full	32.9	33.4	36.9	48.3	41.3	58.2
60% full	14.5	17.1	19.8	31.0	27.7	41.2
40% full	20.5	22.6	25.8	39.0	33.5	38.0
20% full	32.8	18.9	5.2	39.0	34.8	6.4

As can be seen the Percent Leakage at 20% full is rather erratic. This is due to the fact that at low capacities tube location and spacing are the more effective factors than tube size.

Figures 11 and 14 show the Percent Leakage versus Tank Weight for the Bladder Tube (with connectors) and the Box Tube designs respectively, which are considered the best designs using horizontal tubes. There appears to be a slight advantage in favor of the bladder tubes with connectors, as far as leakage protection and weight are concerned. However, it is felt that assembly of the tank and replacement of damaged units would be very much easier and faster if the box tube design were adopted. The optimum tube size in both cases seems to lie somewhere between 4" O.D. and 6" O.D.

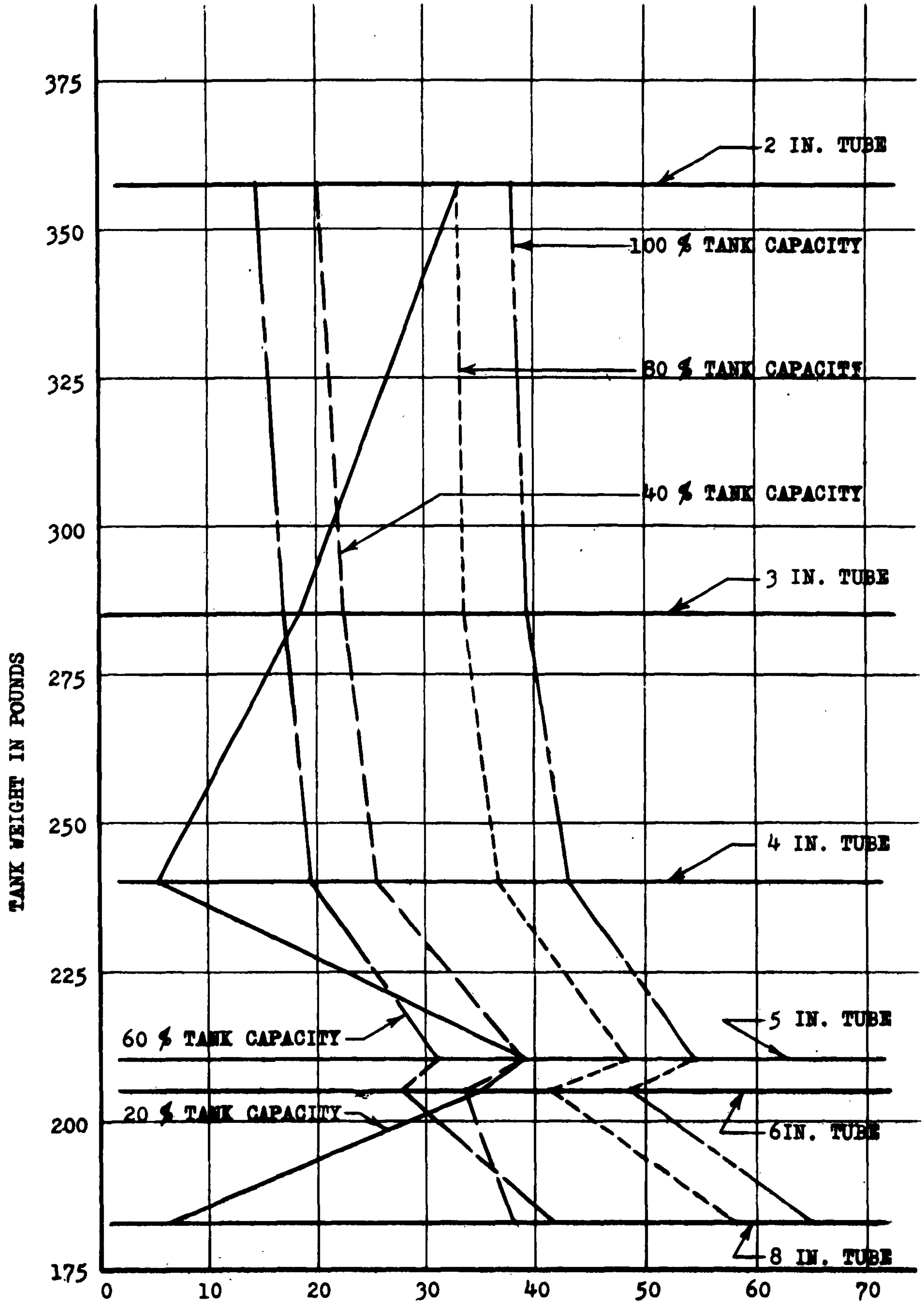


FIG. 14 . PERCENT LEAKAGE VERSUS TANK WEIGHT FOR BOX TUBE.

VERTICAL TUBE TANK

The first design of a vertical tube arrangement used flexible light-weight square tubes suspended from a framework of aluminum tubing (See Fig. 15). The load on this framework was transferred to three aluminum I-beams which were connected by tie brackets in such a manner that the total load could be transferred to the tank mounting brackets on the sides of the tank. It was decided to have four external compartments each fitted with a check valve so as to minimize the loss of fuel from the external compartments. Allowing 10 in. from the sump gives a length of 28 in. for each compartment, of which 27 in. was allowed for tubes and 1 in. for structural members. Fifty-four inches of the fifty-six inches of tank width was allotted to the tubes. Since one of the three I-beams was to be located in the center of the tank, it was necessary to have an even number of tubes across the tank. The edge dimensions of square tubes which gave a whole number of tubes lengthwise and an even number of tubes across the tank were 3.37, 4.50, 6.75, 9.00 and 13.50 in.

The size of aluminum tubing to be used in the framework was calculated by considering the tubes as simple beams with unevenly distributed loads. The tube sizes varied from 1/4 in. OD x 1/32 in. thick for the tank using 3.37 in. square tubes to 1 in. OD x 1/8 in. thick for the tank using 13.5 in. square tubes.

Next the I-beam sizes were calculated and found to be standard 3 in. I sections with a 0.17 in. web for all three beams. For the tie bracket a 3 in. I section with a 0.25 in. web was used for the horizontal member and a 2 in. OD x 1/8 in. thick tube was used for the vertical member. The tank shell and compartment separators were the same as used in the horizontal tube designs.

Table 7 shows the Tank Weight, Numbers of Tubes per Tank and the Percent Leakage at various capacities for the five sizes.

TABLE 7
% LEAKAGE-VERTICAL TUBE TANKS INITIAL DESIGN

Tube Size Inches	No. of Tubes	Vol. per Tube-gal	Weight lbs.	Percent Leakage				
				20	40	60	80	100
3.37 x 3.37	512	1.10	367	0	0	0.4	3.8	14.0
4.5 x 4.5	288	1.97	328	0	0	3.3	6.5	16.7
6.75 x 6.75	128	4.44	286	0	1.1	5.8	12.7	22.1
9 x 9	72	7.89	270	0	1.9	6.0	13.6	24.5
13.5 x 13.5	32	17.70	241	0	3.8	9.0	20.4	31.6

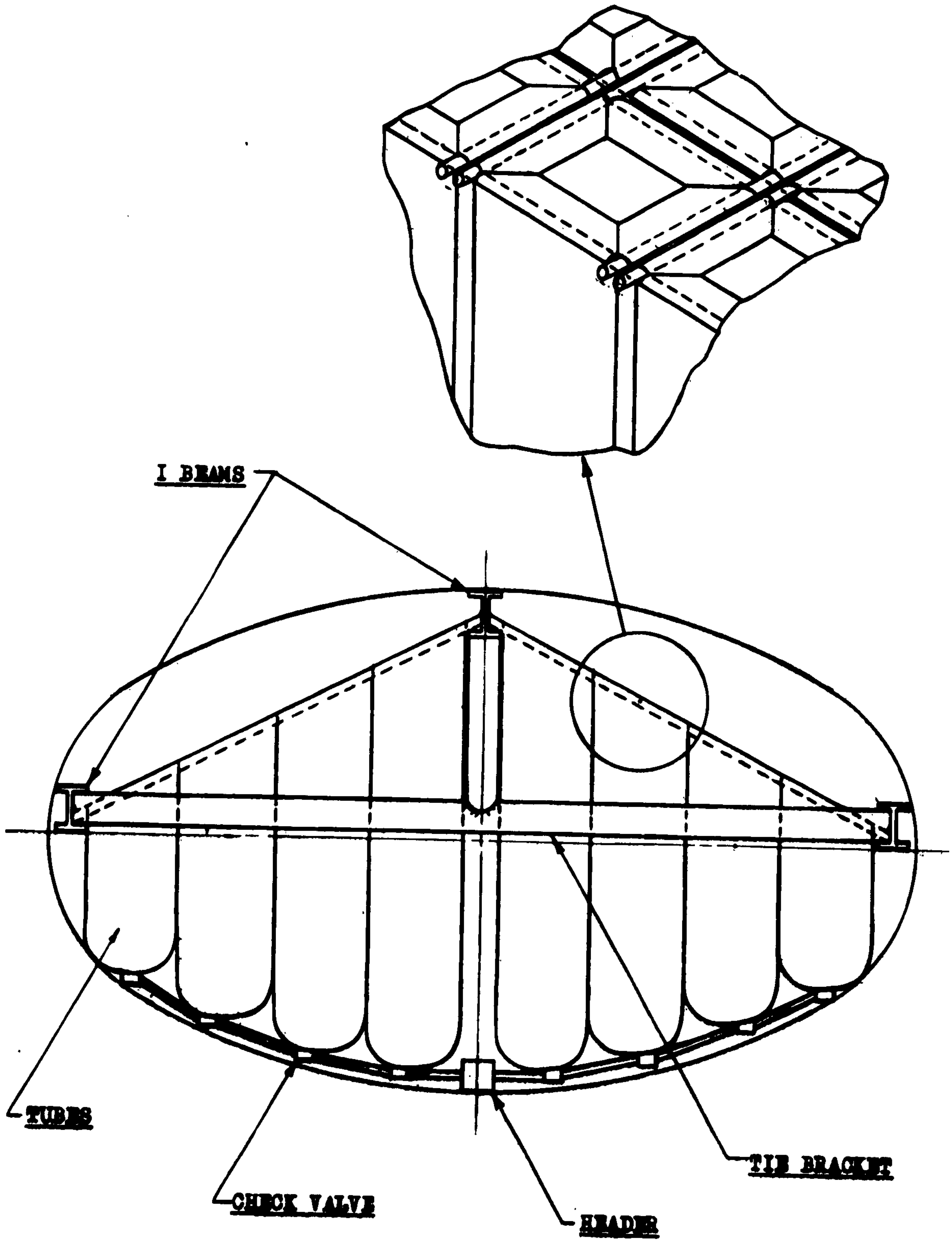


FIG. 15 SUSPENDED VERTICAL TUBE TANK

Evaluation of Leakage Protection was performed in the same manner as was indicated in the section on evaluation of horizontal tube tanks, except that the tubes were assumed to empty simultaneously.

Fig. 16 shows a plot of Percent Leakage versus Tank Weight for the vertical tube tanks. The leakage for all tanks at 20% capacity was zero due, primarily, to the standard shot pattern employed to evaluate the leakage. Although there is no sharp break in the curves, it appears that about a 6 in. tube is the optimum size.

A comparison of the leakage protection of the vertical tube tanks with that of the horizontal box tube tanks revealed the marked advantage of the vertical tubes.

The question of whether there was an advantage for round tubes over square when the tank was only partly filled was answered by calculating the Percent Leakage for a 7-1/4 in. OD tube and 7-1/4 in. x 7-1/4 in. square tube. Since it was desirable to find out if the tube size was also a factor in preference of round or square tubes, a 4 in. OD tube and a 4 in. x 4 in. square tube were also studied.

Previous studies had indicated that when the tubes were placed side by side there was the possibility of damage due to the hydraulic ram effect. This effect, which may be thought of as a shock wave transmitted in all directions when a tube is struck by a projectile, could cause tubes adjacent to the punctured tube to be ruptured. A clearance space of 3/16 in. was thought to be large enought to minimize the effect of the hydraulic ram. Accordingly, the above-mentioned studies of round and square tubes were made with no clearance between tubes and with a 3/16 in. clearance between tubes.

In the vertical tube design a great percentage of the fuel is in the tubes and the necessity of having the external volume compartmented is questionable. To determine the reduction of leakage due to compartmenting, the above studies were made with one external compartment and with three external compartments.

Table 8 shows the Percent Leakage for the various conditions listed above.

Fig. 17 shows the Percent Leakage versus Percent Capacity for the two sizes of tubes both round and square with no clearance between tubes. Fig. 18 shows the same variables for the two sizes of tubes both round and square when a 3/16 in. clearance space existed between the tubes. Both figures indicate that from the leakage protection standpoint, square tubes are superior to the round tubes when the tank is filled to 70% capacity or more. Below 70% capacity the leakage is very small

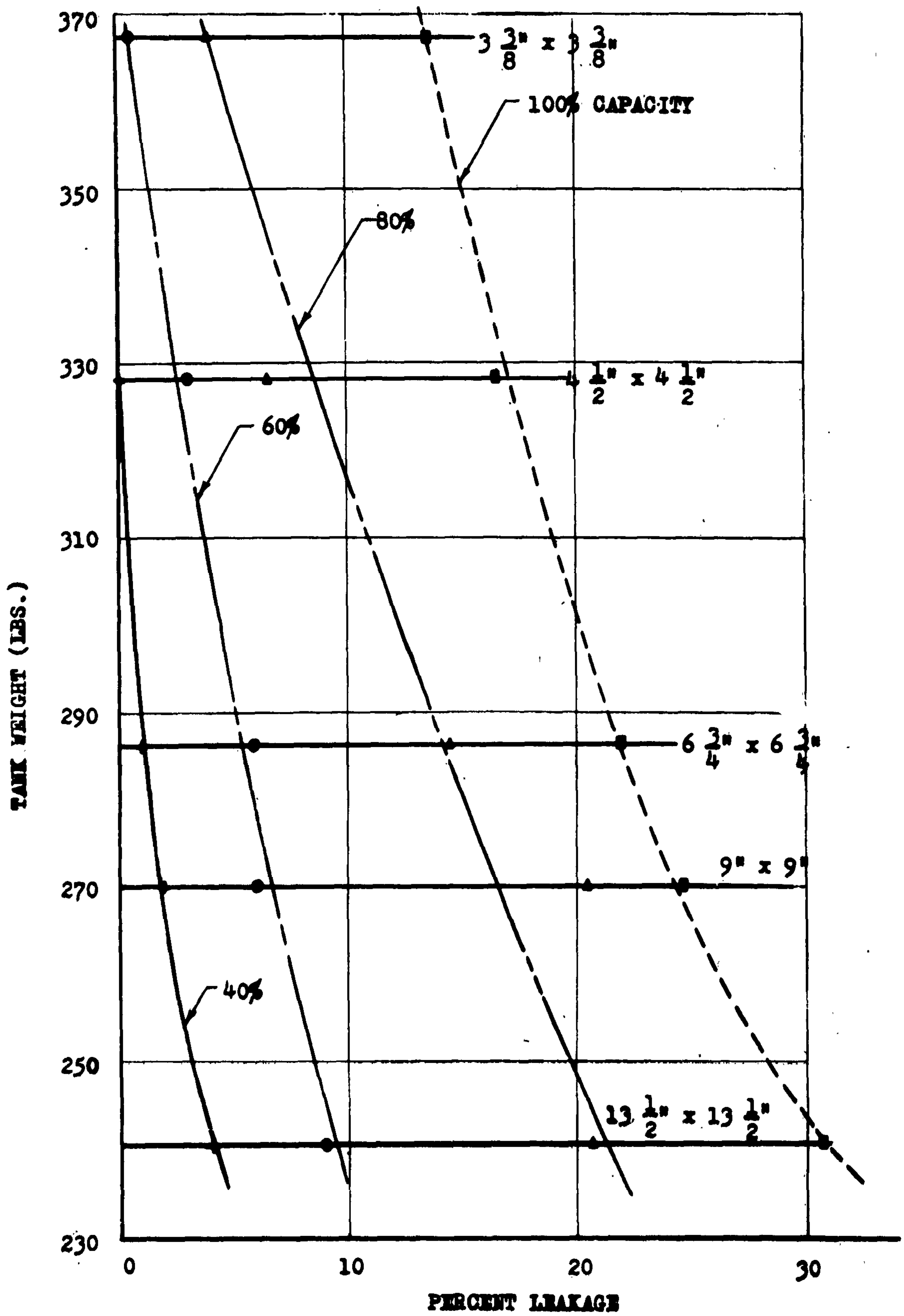


FIG. 16 % LEAKAGE VS. TANK WEIGHT FOR VERTICAL TUBES

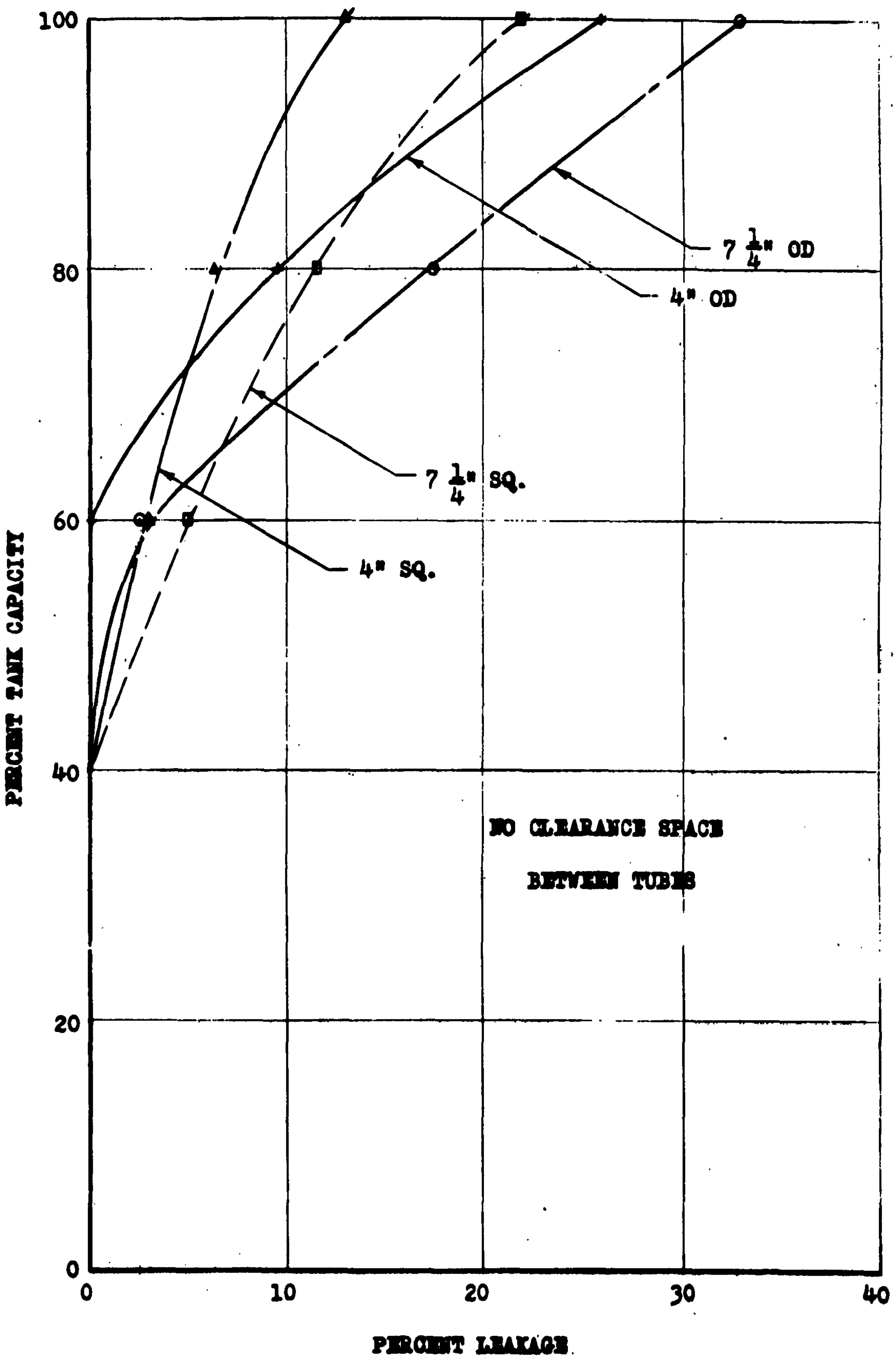


FIG. 17 % LEAKAGE VS. CAPACITY - VERTICAL TUBES (NO CLEARANCE)

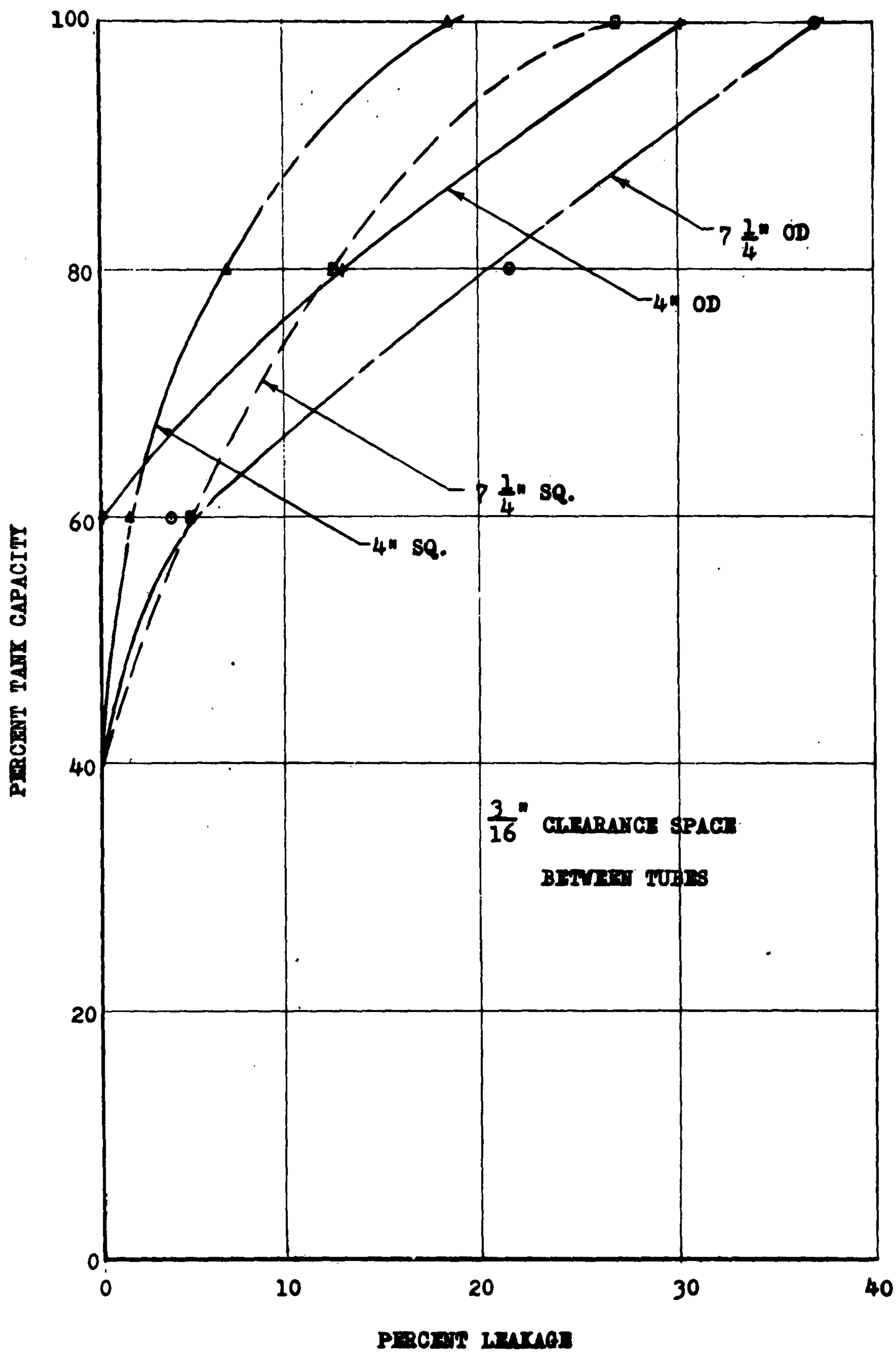


FIG. 18 % LEAKAGE VS. CAPACITY - VERTICAL TUBES (WITH CLEARANCE)

for both round and square tubes. Although it is recognized that the weight of material for the square tubes would be 12.7% greater than the weight of the material for the round tubes, it seems that the increased protection of the square tube more than offsets this weight factor. By comparing the two figures, it can be seen that the clearance space between tubes causes a slight increase in leakage at 80% and 100% capacity, but has practically no effect at lower percent capacities. Reference to Table 8 reveals that compartmenting the external volume results in only a very slight increase in protection.

The conclusion from this last study may be summarized as follows:

1. Square tubes are more desirable than round tubes.
2. A small clearance space between tubes does not decrease the leakage protection significantly.
3. Compartmenting the external volume does not increase the leakage protection significantly.

An analysis of the weight breakdown of the previously described vertical tube tank revealed that about 60% of the tank weight was necessary for supporting the tubes in the tank. In an effort to reduce this component of the total weight, rigid tube materials were investigated as outlined in Section III. The conclusion of this investigation was that a light weight tube material could be fabricated so as to support its own weight and at the same time withstand the bursting pressure created by the fuel. In order to eliminate the supporting or suspending framework, the tubes were designed to sit on the bottom of the tank. This design permitted the load to be evenly distributed across the bottom of the tank and thus transmitted to the cradle or backing board upon which the tank rested. In place of the bladder material shell used in previous designs, a 20-gage aluminum sheet was used. Maintenance of tank cross sectional shape was accomplished by having four 18-gage punched aluminum bulkheads in addition to the two ends. The tube size was selected by considering the previous investigation as to optimum tube size and by calculating the size that would best fit the tank dimension. This resulted in a 6-3/8 in. x 6-3/8 in. tube with a 3/16 in. clearance between tubes.

Table 9 shows the Percent Leakage at various capacities for the second design of vertical tube tank.

TABLE 9

PERCENT LEAKAGE-VERTICAL TUBE TANK-FINAL DESIGN

Percent Capacity	20	40	60	80	100
Percent Leakage	0	0	12.5	15.4	24.3

The approximate weight of this tank was 200 pounds.

TABLE 8

PERCENT LEAKAGE VERTICAL TUBES

Tube Type & Size	Number of Tubes	Gal. per Tube	Gal. in Tubes	Gal. per Ext. Comp.	Number of External Compartments	% LEAKAGE 20% Full	40% Full	60% Full	80% Full	100% Full
NO SPACE BETWEEN TUBES										
4" OD Rd.	377	1.28	482	158	1	0	0	0	9.6	26.2
4" OD Rd.	377	1.28	482	54-50-54	3	0	0	1.7	4.9	23.5
4" Sq.	377	1.63	615	25	1	0	0	3.0	6.6	12.7
4" Sq.	377	1.63	615	8.6-7.8-8.6	3	0	0.8	2.6	6.0	12.3
7-1/4" OD Rd.	112	4.02	450	190	1	0	0	2.6	17.2	33.5
7-1/4" OD Rd.	112	4.02	450	59-72-59	3	0	0	2.7	13.2	30.0
7-1/4" Sq.	112	5.12	573	67	1	0	0	4.9	11.5	22.2
7-1/4" Sq.	112	5.12	573	21-25-21	3	0	0.9	3.5	9.5	21.0
3/16" SPACE BETWEEN TUBES										
4" OD Rd.	336	1.32	443	197	1	0	0	0	13.2	30.5
4" OD Rd.	336	1.32	443	63-71-63	3	0	0	1.0	9.2	26.9
4" Sq.	336	1.68	565	75	1	0	0	1.3	6.4	18.3
4" Sq.	336	1.68	565	24-27-24	3	0	0	2.0	4.6	16.9
7-1/4" OD Rd.	105	3.99	419	221	1	0	0	3.5	21.3	36.9
7-1/4" OD Rd.	105	3.99	419	74-74-74	3	0	0	3.6	17.0	33.2
7-1/4" Sq.	105	5.07	532	108	1	0	0	4.2	12.5	26.8
7-1/4" Sq.	105	5.07	532	36-36-36	3	0	0	3.9	9.4	24.9

SECTION III

FINAL DESIGN

MATERIAL:

In the designs which were used as the basis of the weight analysis for the evaluation studies as reported in Section II, it was assumed that the compartments would be made of bladder material with a weight of 0.06 lb/sq.ft. Further, the compartments of the cellular tanks were to be supported on a structural member made of square aluminum tubing. After giving more attention to the details of such a design, it was felt that this method of construction was not practical because of the difficulties of fabrication. Further, it was felt that the uncertainties associated with the support of the bladder cell on its bottom, particularly under the high g loading, necessitated a more elaborate method of support or the selection of a self-supporting material. Also, the bladder material, if used in the vertical tube tanks, would not be self supporting, thus requiring a rather heavy structural element to support the tubes from the top and transfer the load to the bottom of the tank. Hence, a search was made for a better material, both from a strength and weight standpoint, with the hope that some material or combination of materials could be found which would have a weight as low as that assumed for the bladder material (0.06 lbs/sq.ft.)

As a first choice, it was thought that a wire mesh screen impregnated with plastic might have possibilities. Thus several different wire meshes were investigated. Following are the calculations for weight per unit area of these meshes:

1. Consider 20 mesh wire cloth (wire size 23-34 Roebling). In one square inch there are 40 inches of wire. For 30 gage (0.014),

$$\text{Weight} = \frac{(3.14)(0.014)^2}{4}\frac{(40)(490)}{(1728)} = 0.00174 \text{ lbs/in}^2$$

2. Consider 10 mesh wire cloth (wire size 18-29 Roebling) In one square inch there are 20 inches of wire. For 29 gage (0.015),

$$\text{Weight} = \frac{(3.14)(0.015)^2}{4}\frac{(20)(490)}{(1728)} = 0.0010 \text{ lbs/in}^2$$

3. Consider 5 mesh wire cloth (wire size 13-24 Roebling). In one square inch there are 10 inches of wire. For 24 gage (0.023),

$$\text{Weight} = \frac{(3.14)(0.023)^2(10)(490)}{(4)\ (1728)} = 0.00118 \text{ lbs/in}^2$$

These three meshes have weights of 0.25, 0.14, and 0.17 lbs/sq.ft. These values are considerably higher than the weight of bladder material and are for the bare wire alone. When plastic or other material is added to the structure, the weight will be higher. Hence, it seemed that a further investigation of wire mesh would not be warranted.

Another construction that was investigated was a lamination, consisting of a layer of aluminum foil on the inside with layers of fiberglas on the outside, bonded together with a plastic. Such a construction has quite a bit of rigidity, but the unit weight of 0.164 lbs/sq.ft. was over twice the weight of the bladder material. This construction was not considered further.

Still another possibility would be to form the cells of a plastic, which would be self supporting. For a trial, tubes of 1/16 in. wall thickness were considered. A stress analysis revealed that this thickness was necessary under the high loading. However, the unit weight was around 0.32 lbs/sq.ft. for a specific gravity of 1.0. Again, this appears to be quite high and would result in an excessive tank weight.

An investigation for light-weight materials to be used for structural and semi-structural purposes has been under way for some time by the Aircraft Industries Association of America, Inc. One such scheme was the use of a honeycomb of aluminum foil. (See Aircraft Technical Committee Report No. ARTC-5). Such a construction could not be used in making thin walls for structural purposes, due to the low strength in an axial direction. However, Fig. 21 shows details of a new design employing corrugated aluminum foil which does have high strength in an axial direction. The aluminum foil has a thickness of 0.001 in. and the corrugations are 0.015 in. spaced on 0.0473 in. centers. Each layer is criss-crossed at 90° to the other. The total sheet is formed of five layers, as described, with outside layers of non-corrugated foil. The total thickness is 0.077 in. and the weight is 0.127 lb. per sq.ft. As this weight compares favorably with the other materials, it seems likely that corrugated aluminum foil has possibilities as a structural element.

A strength analysis reveals that the stresses would not be excessive on the bottom of a cell filled with fuel under 7.33 g loading, when the cell is supported on opposite edges. It also seems that such a light-weight material might be very useful in other structural elements used in aircraft. It should be pointed out, however, that the method of producing such a light-weight sheet has not been developed and some research will have to be done to find such a method. Such things as the formation of corrugations and the bonding of the sheets need to be considered. As an example, it may be possible to bond the sheets by resistance welding or by cementing with an adhesive. Other considerations, such as the forming of the compartments will also need investigation.

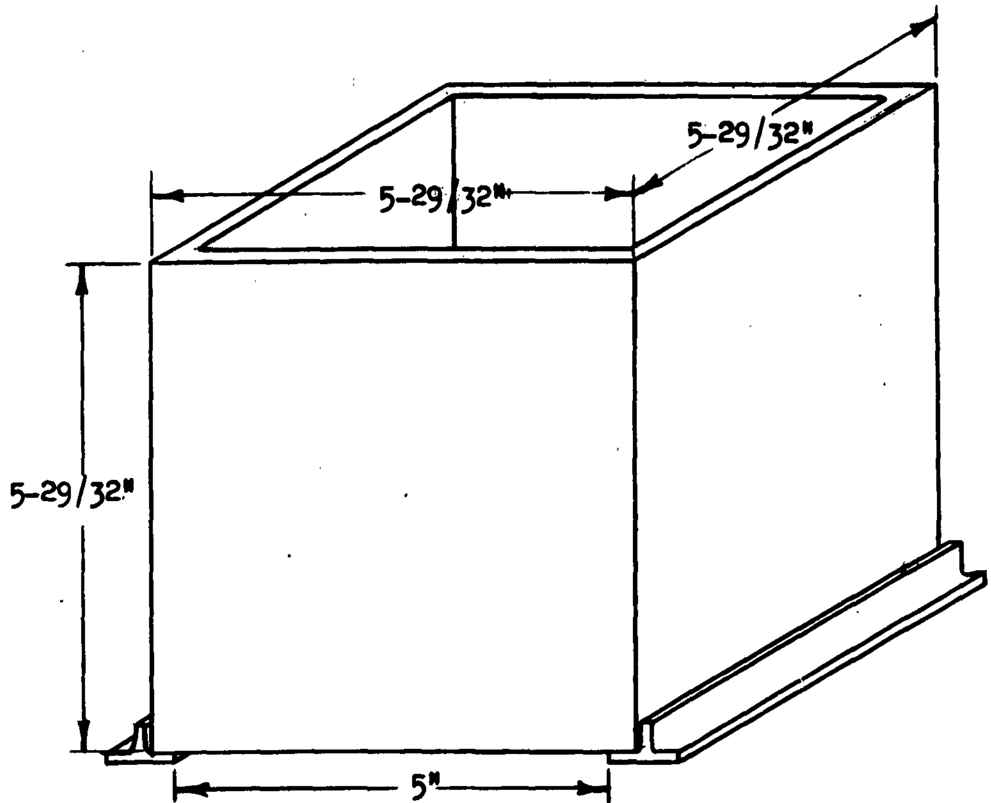


FIG. 19 FUEL CELL

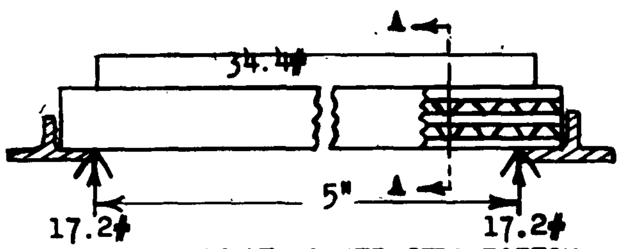
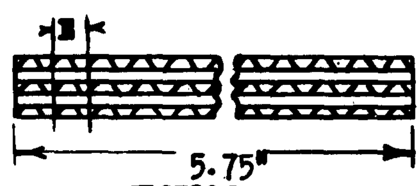


FIG.20 END VIEW OF LOADED CELL BOTTOM



SECTION A-A

FIG. 21 CELL BOTTOM CROSS SECTION

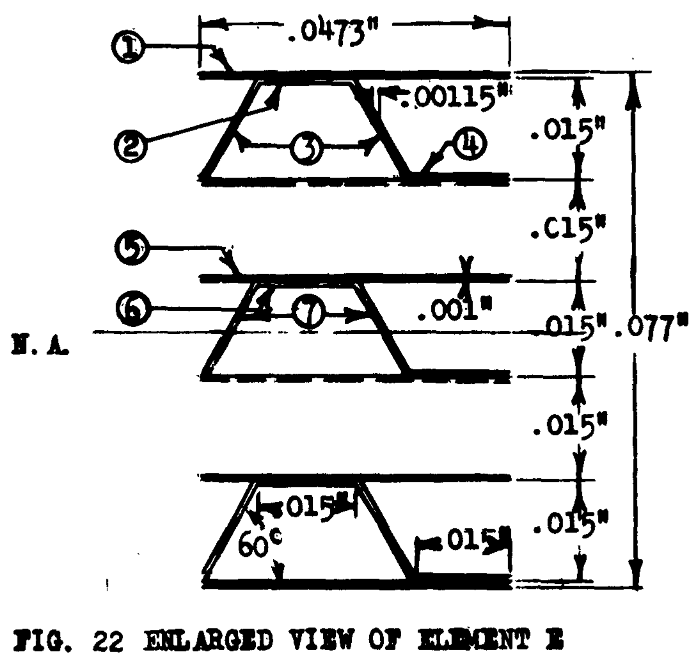


FIG. 22 ENLARGED VIEW OF ELEMENT E

However, the chances appear very good that solutions to these problems may be found and that this method of construction will result in a very light-weight compartmented fuel tank. For this reason, it was decided to base the final design of both the cellular and tube tanks on the use of corrugated aluminum foil as the material for the compartments.

CELLULAR TANKS:

The final design of the cellular tank is shown on Aeronautical Research Laboratory Drawing No. 6049 "Assembly-Cellular Tank." A description of the design follows:
(1) OUTER SHELL: The outer shell is made of 20-gage (B & S) (0.0320 in.) aluminum sheet. It is formed in two halves, being split on a horizontal plane dividing the vertical height of the tank. The two parts, upper and lower, are formed of aluminum sheets 2 ft. wide, and welded together with external seams. The heads are made by stamping and have internal, integrally formed ribs for stiffening. These heads are welded to the upper and lower parts of the tank shell. The two shells are fastened together by means of a bolting flange, 1.00 in. wide with a 0.03 in. gasket between. Aluminum bolts on 3-in. centers are used as a fastening means. The whole tank rests on a bottom board of such a contour that full support for the internal structure is obtained through the outer shell. Bumper blocks placed along the sides and top transfer the acceleration loads to the airframe or other supporting structure. The outer shell has a sump centrally located, which is bolted on from below. This sump extends some 3.50 in. below the contour of the outer shell. Also, vent, filler and drain caps are provided. It is estimated that the weight of the outer shell, without the sump, is 62 pounds.
(2) INTERNAL STRUCTURE: To provide a support for the cells, an internal structure has been designed consisting of a framework of such a nature that the cells and their integral plumbing can be slipped in place, like drawers in a filing cabinet. This framework consists of long T sections of extruded aluminum which are spaced apart by both vertical and horizontal aluminum bars, all welded together into an integral unit. Space below the framework is provided for the lines connecting the cells to the sump. A considerable portion of the tank weight is due to this frame, its weight being estimated at 125 lbs.

This structure is so shaped that it fits inside the outer shell and forms a supporting member thereby. It then transfers the load of the fuel directly to the bottom support board or endwise through bumper blocks to the airframe.
(3) CELLS: The cells are cubical, open at the top and made of corrugated aluminum foil. From the Percent Leakage vs Weight Curves for the cellular tank, it was determined that a proper cell dimension would be 6 in. on the side. This results in

about 780 cells with an estimated weight of 110 lbs. The following calculations were made to determine the stresses set up in the cells and frame under the specified accelerated loads.

Stress Analysis of Cell Bottom:

The fuel cells are supported on two T-sections as shown in Fig. 20. The cell walls and bottom are built up of corrugated sheets of 0.001 in. thick aluminum foil as shown in Figs. 19, 20, 21, and 22, on page 40.

The weight of fuel in each cell may be calculated in the following manner:

Wall thickness = .077 in.

$\therefore$ Inside cell dimensions = 5.906 -2(.077) = 5.75 in.

Inside cell volume = (5.75 in.)^3 = 190 in.^{3} = 0.113 ft.^{3}

Capacity of cell (gallons)

= (0.113) (7.48) = 0.82 gals.

$\therefore$ Weight of fuel in cell at 7g

$= .82 \text{ gals } \left(\frac{6 \text{ lb.}}{\text{gal.}}\right) 7 = 34.4 \text{ lb.}$

A stress analysis of the cell bottom may be carried out by assuming the bottom to be a simple beam supported on knife edges 5 in. apart and carrying a uniformly distributed load of 34.4 lb. (See Fig. 20). The maximum bending stress is assumed to be the greatest stress in the bottom and may be calculated by the well-known formula:

$$S = \frac{MC}{I}$$

where

S = maximum bending stress in beam

M = bending moment at midpoint of beam

C = distance of outermost portion of beam from neutral axis (N.A.) of the beam

I = moment of inertia of beam cross section about its neutral axis.

Calculation of moment of inertia of beam cross section about its neutral axis may most easily be accomplished by considering a typical element of the cross section (see Fig. 22). The moment of inertia of this element about the neutral axis may by found by calculating and summing the moments of inertia of the element's component members about the neutral axis. The moment of inertia of the element may then be multiplied by the number of elements composing the entire beam cross section to give the moment of inertia of the entire beam cross section about the neutral axis.

Moments of inertia of the component members of the typical element of the beam cross section are calculated by the

formula

$$I_{N.A.} = I_{x_o} + Ad^2$$

where $I_{N.A.}$ = moment of inertia of member about neutral axis of beam

I_{x_o} = moment of inertia of member about its geometrical axis.

A = cross sectional area of member

d = distance of geometrical axis of member from neutral axis of beam

For members parallel to the neutral axis the term I_{x_o} is negligible and was not considered in calculating the moment of inertia of the beam.

Following are the numerical calculations for the moment of inertia. The numbers refer to Fig. 22.

(1) $I_{N.A.}$ of outerskin (member 1)
$= (4.73 \times 10^{-2})\ (10^{-3})(3.8 \times 10^{-2})^2$
$= 68.4 \times 10^{-9}$ in.^{4}

(2) $I_{N.A.}$ of outer corrugation top (member 2)
$= (1.5 \times 10^{-2})(10^{-3})(3.7 \times 10^{-2})^2$
$= 20.6 \times 10^{-9}$ in.^{4}

(3) $I_{N.A.}$ of outer corrugation diagonals (member 3)
$= 2\ \ (.038)(1.15 \times 10^{-3})(1.5 \times 10^{-2})^3$
$+ (1.15 \times 10^{-3})(1.5 \times 10^{-2})(3 \times 10^{-2})^2$
$= 2(.322 \times 10^{-9}) = 31.6 \times 10^{-9}$ in.^{4}

(4) $I_{N.A.}$ of outer corrugation bottom (member 4)
$= (1.5 \times 10^{-2})(10^{-3})(2.3 \times 10^{-2})^2$
$= (7.94 \times 10^{-9}$ in.^{4}

(5) $I_{N.A.}$ of middle corrugation bottom (member 5)
$= (4.73 \times 10^{-2})(10^{-3})(8 \times 10^{-3})^2$
$= 3.02 \times 10^{-9}$ in.^{4}

(6) $I_{N.A.}$ of inner corrugation top (member 6)
$= (1.5 \times 10^{-2})(10^{-3})(7 \times 10^{-3})^2$
$= .735 \times 10^{-9}$in.^{4}

(7) $I_{N.A.}$ of inner corrugation diagonals (member 7) is negligible.

∴ Moment of inertia of entire element about neutral axis
$= 2\ \ (68.4 + 20.6 + 31.6 + 7.94 + 3.02 + 7.35)$
$= \times 10^{-9} = 264.6 \times 10^{-9}$ in.^{4}

Beam cross section is composed of $\frac{5.75}{.0473} = 121.5$ typical elements.

∴Moment of inertia of beam cross section about neutral axis
$= (121.5)(264.6 \times 10^{-9}\ \text{in.}^4) = 3.21 \times 10^{-5}$ in.^{4}

$$S = \frac{MC}{I} \quad 17.2(2.4)-17.2(1.25)\ (.0385)$$

$$= \frac{(17.2)\ (1.25)\ (.0385)}{3.21 \times 10^{-5}} = 25{,}800 \text{ psi}$$

The actual maximum stress in the cell bottom would be much less than this value because of the following conditions which were neglected in the stress analysis:

1. The cell walls would help support the bottom, thus reducing the stress in the bottom.
2. The ends of the beam shown in Fig. 20 would not be entirely free to turn about the knife edges but would be fixed to some extent by the weight of the fuel on the overhanging parts of the beam. This would reduce the bending stress at the middle of the beam.

Consequently, it is expected that such a construction is entirely practical with aluminum foil as the material.

Stress Analysis of T-Beams:

Another calculation of interest is the stress in the T-sections due to the load of the cells. Following are the calculations for this stress:

The maximum span of each T-beam is 30 in. between supports. On this 30-in. span each beam supports 5 fuel cells which, when full, and at 7g, weigh 34.4 lbs. each (see Stress Analysis of Cell Bottom). Therefore each T-beam was considered to be supported on knife-edges at both ends and to carry a uniformly distributed load of (5 x 34.4 = 172 lbs) (see Fig. 23)

Maximum bending stress in any T-beam may be calculated by means of the formula

$$S = \frac{M}{Z}$$

where S = maximum bending stress in beam
M = bending moment at midpoint of beam
Z = section modulus of beam

The T-beams used were selected from the Alcoa Structural Handbook. Cross sectional dimensions are shown in Fig. 24. The section modulus of this cross section about its neutral axis (N.A.) is given by the Alcoa Handbook as 0.032 in.^{3}

Then $$S = \frac{86(7.5 \text{ lb-in.}}{.032 \text{ in}^3} = 20{,}200 \text{ psi.}$$

14S-0 aluminum extrusions have a tensile strength of 35,000 psi; therefore maximum stress on the cellular tank T-beams is within acceptable limits.

(4) <u>DRAIN LINES</u>: Drain lines from each cell permit the fuel in the cell to be conducted to the sump without danger of leakage into any other cell. This is accomplished by providing

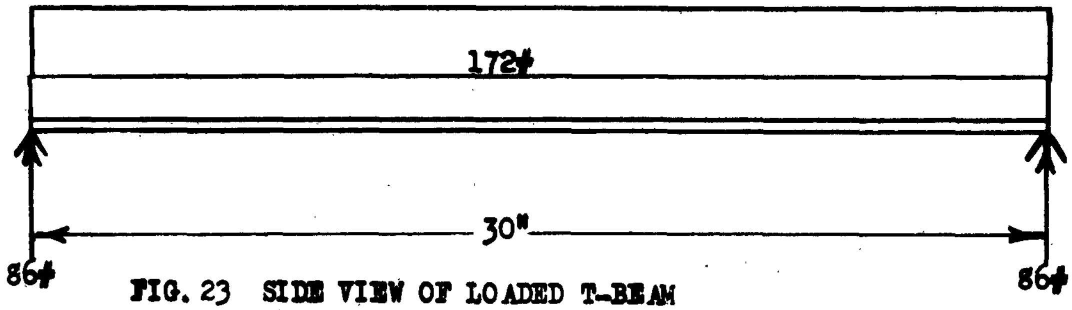


FIG. 23 SIDE VIEW OF LOADED T-BEAM

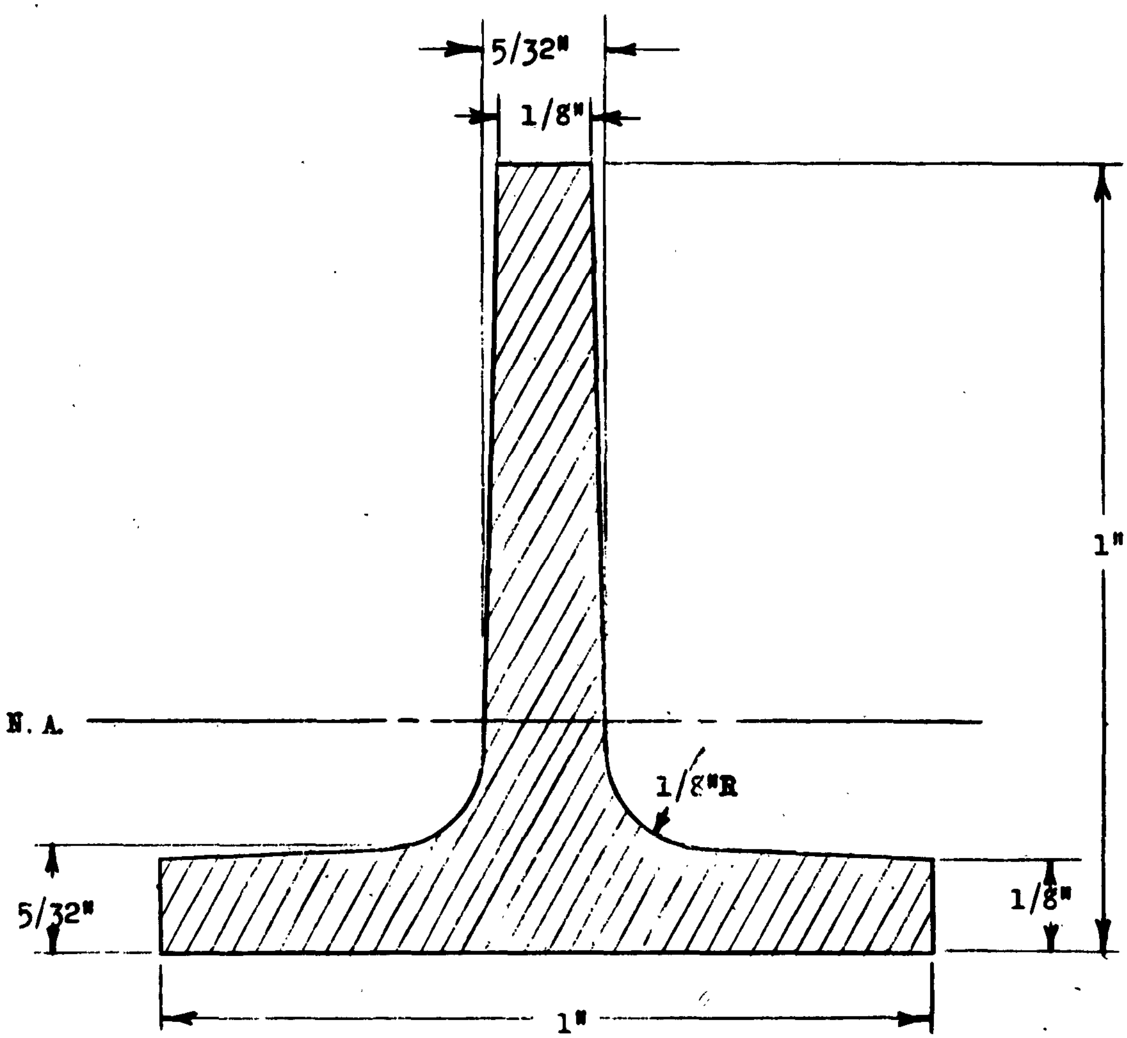


FIG. 24 T-BEAM CROSS SECTION

each cell with a check valve. In the event any cell becomes damaged, the check valve would prevent any backflow from cells which might be elevated over the damaged cell. These check valves are shown in Aeronautical Research Laboratory Drawing No. 4238 "Valve-Check--Cellular Tank." The valve is made with a die-cast aluminum body and provides means of attaching plastic tubes which connect the cells of one row in parallel. All rows of cells in any vertical plane are connected on the end by a vertical tube resting in the indentations formed in the heads of the outer shell. Each group is then connected by a header which in turn connects with a large drain line which lies on the bottom of the tank. This line runs the length of the tank and empties into a collector. The only outlet for the collector is through the float valve into the sump. Hence, each cell is individually connected to the sump by this float valve and serves to keep the sump supplied with fuel.

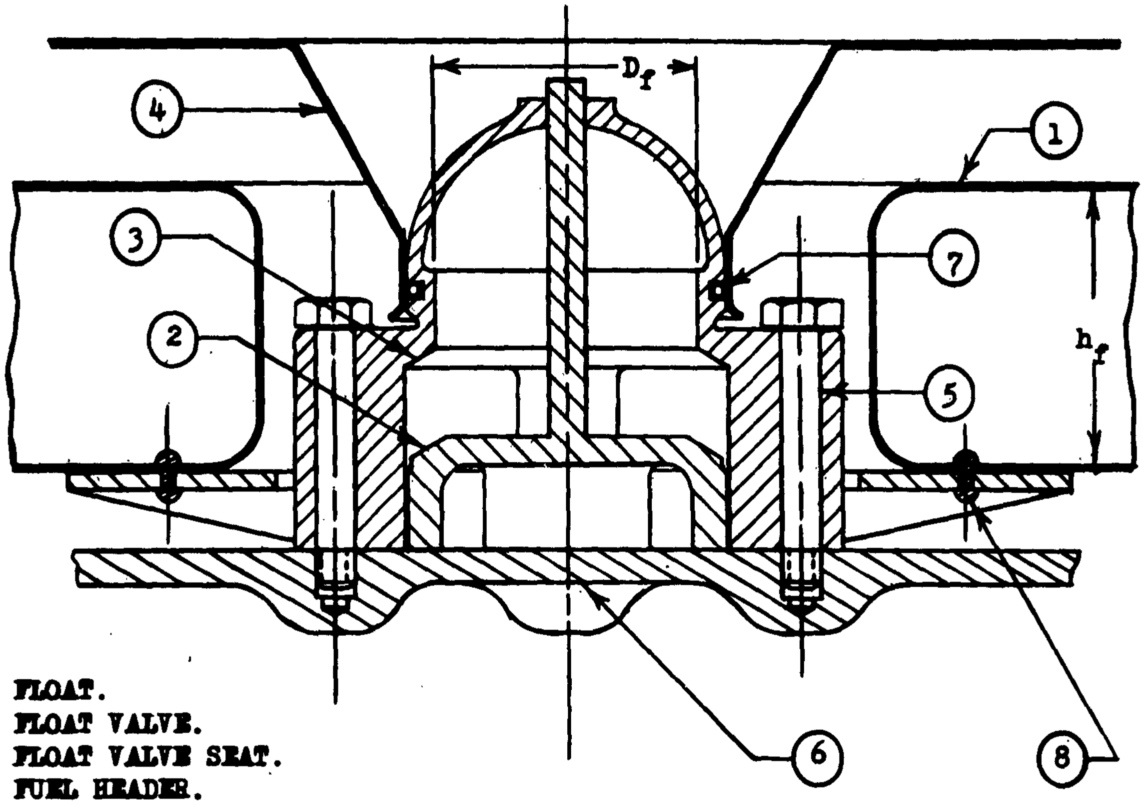
The estimated weight of the plastic drain tubes is 16 lbs.

(5) FLOAT VALVE: The float valve regulates the flow from the collector to the sump. All external space will drain into the sump which contains an internally mounted booster pump. The sump sits a little below the bottom board and has internal openings to the tank. As the level drops in the sump, the float valve opens, thus connecting the cells directly to the sump. This arrangement permits each cell to be connected to the sump, while at the same time allowing the outer space to to be emptied first. The estimated weight of the pump, float valve, sump and drain header is 15 lbs..

It is required that the external volume of the tank (which is the volume between the cells) be emptied before any fuel is taken from the cells. Therefore, it is necessary that a float valve, to control flow from the cells, be provided that will be operated by the level of the fuel in the external volume. Such a valve is shown in Figure 25.

A 13-1/2 x 13-1/2 in. sump was designed, extending approximately 3.50 in. below the bottom of the tank. Fuel from the external volume flows freely into the sump, where is located the inlet to the fuel pump. The float valve, which is also located in this sump, is held in the closed position as long as any fuel remains in the external volume. When this fuel is exhausted and the fuel pump lowers the fuel level in the sump, the valve opens and allows gas to flow from the header into the sump. The cells can discharge only through the header.

This valve assembly fastens to the bottom of the sump as shown in Fig. 25. The fuel pump is also fastened to this bottom and the entire unit of pump, valve and sump bottom may



1 FLOAT.
2 FLOAT VALVE.
3 FLOAT VALVE SEAT.
4 FUEL HEADER.
5 THROUGH BOLTS.
6 BOTTOM OF SUMP.
7 O-RING SEAL.
8 RIVET.

FIG.25 . FLOAT VALVE ASSEMBLY.

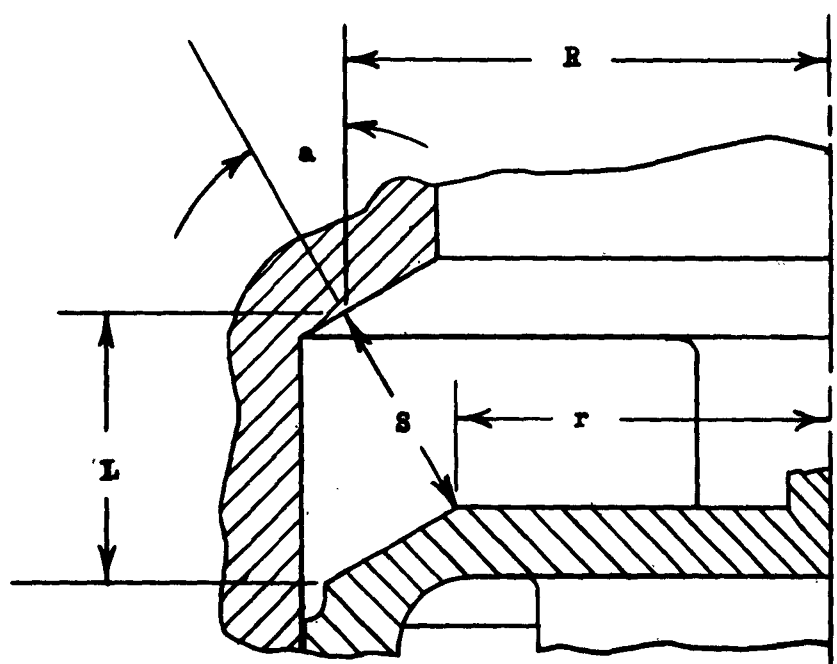


FIG.26 . FREE AREA THROUGH VALVE OPENING.

be removed from the tank simply by removing the bolts holding the bottom to the sump walls. The fuel header shown is fastened permanently to the tank proper. The O-ring seals the header to the valve. This design allows the pump and valve assembly to be completed and tested prior to installation, and permits disassembly for cleaning and repairing with a minimum of time and effort.

The design described above decreases the fuel loss when the tank is damaged due to the fact that when a cell is damaged and its contents are dumped into the external volume, the float valve closes and the fuel pump uses all of this fuel before taking any more from the cells. The entire assembly was designed to have a strength much greater than the normal requirements, since it was felt that in the assembly operation a much greater stress might be imposed.

Assume that the maximum force on the valve comes when the tank is full of gasoline, and that the valve is closed

Assume that

$D_f = 2$ in. $= .16$ ft.

and that

$w = 44.8$ lbs./ft.^{3}

where D_f = diameter of the valve opening as shown in Fig. 25

and w = specific weight of gasoline

then
$$W_g = h_g \, w \, 3.14 \frac{D_f^2}{4} \qquad (1)$$

Where W_g = force of gasoline on valve

h_g = head of gasoline over valve

Assume that the valve is located 3 in. below the bottom of the tank. Since the depth of gasoline in the tank when full is 32 in., then

$$h_g = 32 + 3 = 35 \text{ in.}$$

$$\text{and } W_g = \frac{35 \times 44.8 \times 3.14}{12 \times 4 \times (6)^2} = 2.86 \text{ lbs.}$$

The area of metal in the top of the float (A_f), determined by planimeter measurement, is 92.13 in.^{2}

The sum of top and bottom areas is 184.26 in.^{2}

The total perimeter of the float is 58.4 in. Assume that the float is 3 in. high. Then the area of metal in the sides of the float is 175.2 in.^{2}

Then the total area of metal in the float is 359.5 in.^{2}

The float is made from aluminum of 0.020 in. thickness. The volume of metal in the float is 7.188 in.^{3} Since the specific weight of the aluminum is approximately 0.0975 lb/in.^{3},

then the total weight of the float is 0.701 lbs.

Assume that the weight of the valve that is fastened to the float is 0.5 lbs. An approximate calculation of this weight shows that this assumed value is high by approximately 50% of the assumed value.

Then the total weight of the float (W_f) is given by

$$W_f = 0.70 + 0.50 = 1.20 \text{ lbs.} \qquad (2)$$

The total buoyant force acting on the float (W_B) is equal to the weight of fluid displaced.

Then $\quad W_B = h_f A_f w \qquad (3)$

Where $\quad h_f$ = height of float
and $\quad A_f$ = area of float

The minimum buoyant force must be equal to the sum of the weight of the valve assembly and the force of the fuel on the valve, or

$$W_B = W_g + W_f \qquad (4)$$

The minimum float height necessary to close the valve is given by

$$h_f = \frac{(W_g + W_F)\ 1728}{A_{fw}}$$

Then

$$h_f = \frac{(2.86 + 1.20)\ 1728}{92.13 \times 44.8} = 1.7 \text{ in.}$$

Therefore a float height of 2 in. will be used.
Three assumptions of

(1) Valve 3 in. below bottom of tank.
(2) Float height of 3 in.
(3) Valve weight of 0.5 lbs.

are all too high. Correcting all of these values will decrease h_f by a very small amount. Since the 2 in. used has a 15% safety factor above the minimum, the decrease in h_f required will only increase the safety factor if the value of 2 in. is not changed.

To determine the length of travel of the float (L), assume that the radial area through which fuel can flow past the valve must be greater than the face area of the valve. In Fig. 26, it can be seen that this minimum area is the area of a frustum of a cone having a small radius r, a large radius R, and a slant height S.

This area (A_s) is given by

$$A_s = 3.14\ S(R + r)$$

and $$S = \frac{(R - r)}{\text{Sin } a}$$

Then $$A_S = \frac{(R - r)(R + r)}{\text{Sin } a}$$

The face area of the valve (A_V) is given by
$$A_V = \frac{3.14 \; D^2 f}{4}$$

For $$A_S = A_V$$
$$\frac{3.14 \; D^2 f}{4} = \frac{3.14 \; (R^2 - r^2)}{\text{Sin } a}$$

For this valve,

$r = 15/16$ in.

$a = 30°$

$D_f = 2$ in.

$$\frac{3.14 \; (2)^2}{4} = \frac{3.14 \; (R^2 - r^2)}{\text{Sin } a}$$

or $$R^2 = r^2 + \text{Sin } a$$
$$R^2 = \frac{(15)^2}{(16)^2} + \text{Sin } 30° = 1.38 \text{ in.}^2$$

Therefore,

R minimum = 1.175

But the value of R for this design is:

R - 1.22 in.

This shows that the radial area is greater than the face area of the valve, which was required.

The length of travel of the float is given by

$$L = \frac{S}{\text{Cos } a} = \frac{R - r}{\text{Sin } a \times \text{Cos } a}$$

$$L = \frac{\frac{39}{32} - \frac{30}{32}}{\text{Sin } 30° \times \text{Cos } 30°} = 0.65$$

All of the above calculations are based on an acceleration of g ft./sec^2. For an acceleration of ng ft/sec^2, every weight in these calculations must be divided by g and then multiplied by ng.

This requires that equations (1), (2) and (3) each be multiplied by n. When this is done, equation (4) becomes

$$nW_B = nW_g + nW_f$$

or $$W_B = (W_g + W_f)$$

which is the same as the original equation. Since equation (5) comes directly from this equation, the h_f calculated by equation (5) is correct for any acceleration. This shows that the operation of the float, in respect to the fuel, will not be changed by a change in load factor.

This does not show, however, what position the valve will take when the system is subjected to a negative acceleration. This can be visualized by remembering that for a positive g, the float operates on the surface of the fuel. This is saying that the float will be on the surface of the fuel opposite to the direction of the acceleration. Then when the acceleration becomes negative, the fuel will try to fill the upper portion of the sump, and the float will operate on the bottom surface of the fuel, provided this condition exists long enough to allow the float to change its position in the fuel. This indicates that during a sustained negative acceleration, the valve will open and the fuel will flow from the sump back up to the cells, and uncover the inlet to the fuel pump.

This is the same condition that exists in the fuel tanks in use today.

(6) VENT: The vents on this fuel tank serves two purposes: (1) to allow air to escape from the tank during refueling and (2) to allow air to escape from the tank while the aircraft is climbing so as to prevent an excessive pressure differential between the inside and outside of the tank. Therefore the two factors governing the vent area are the maximum fueling rate and the maximum climbing rate, which are specified for this tank as 400 gallons per minute, and 25,000 feet per minute respectively.

The allowable stress in the tank walls is taken as 3000 psi. The pressure differential between the inside and outside of the tank necessary to cause this stress may be calculated in the following manner:

$$F = Sa$$

Where S = stress in tank walls

F = total force on projected area of largest side.

A = wall cross sectional area resisting force F

Also $P = F/A'$

Where P = pressure differential between inside and outside of the tank

A' = projected area of largest side.

Then (See Fig. 27).

$$P = \frac{SA}{A'} = 3000 \frac{\text{lb}}{\text{in}^2} \frac{[2(122) + 2(56)]\ .032}{122(56)} = 5 \text{ psi}$$

These calculations show that a pressure differential of 5 psi will cause a transverse stress in the tank walls of 3000 psi. This is the maximum stress in the walls since the transverse stress is always twice as great as the longitudinal stress in the walls of a hollow container under pressure.

The vent area necessary to prevent a pressure differential exceeding 5 psi during refueling at 400 gpm will now be calculated

$$\text{Ft}^3 \text{ of fuel in} = \text{ft.}^3 \text{ of air vent} = 400 \frac{\text{gal}}{7.48 \text{ gal}}$$

$$= 53.5 \frac{\text{ft}^3}{\text{min.}}$$

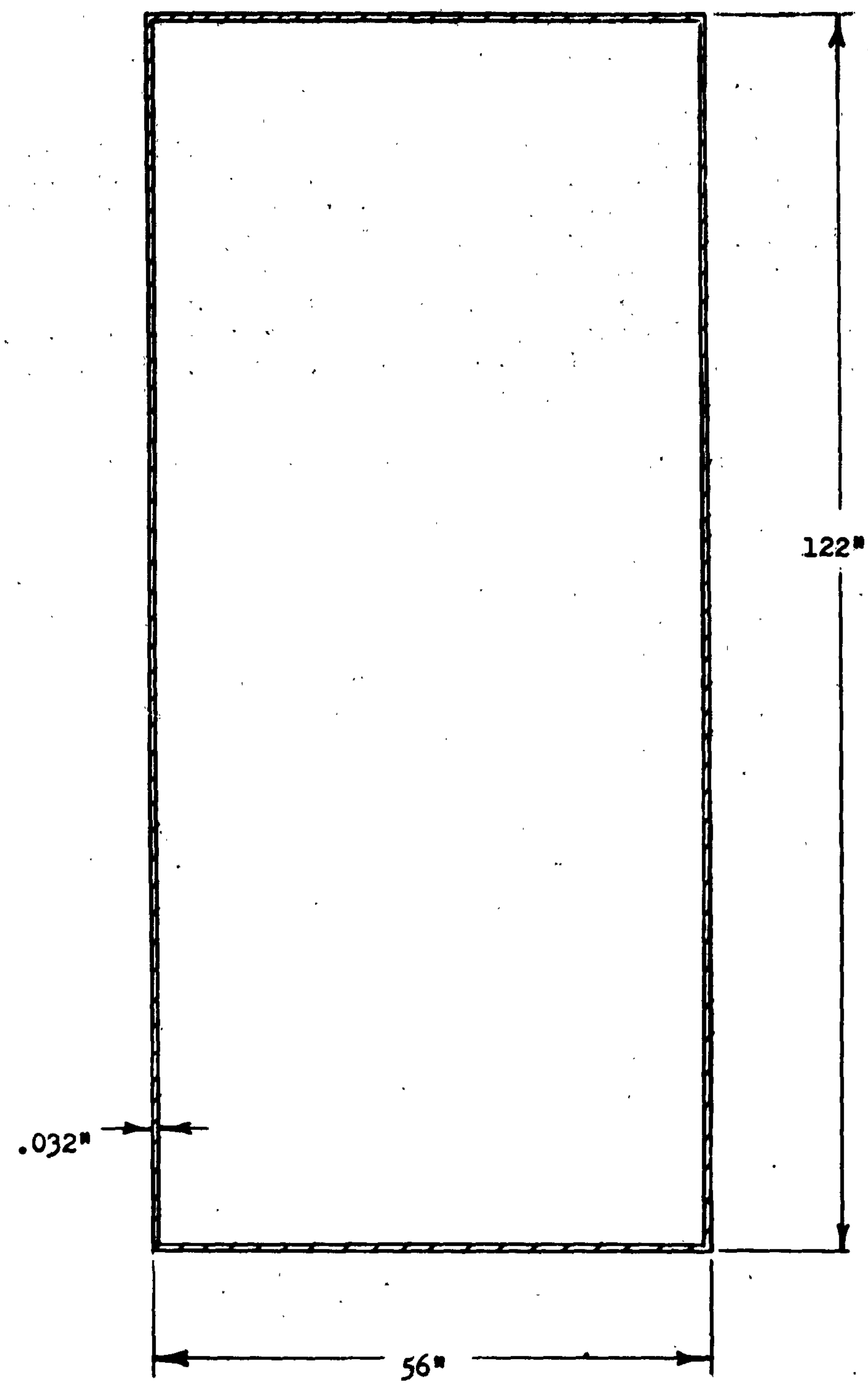


FIG. 27 LONGITUDINAL SECTION OF OUTER TANK SHELL

The vent area may be calculated by the equation

$$A = \frac{Q}{V} = \frac{Q}{c\sqrt{2gP/w}}$$

Where
A = Vent area
Q = volume of air discharged per minute
c = discharge coefficient of vent opening
g = gravitational constant
h = pressure head on air in tank
p = pressure differential = gage pressure
w = specific weight of air at absolute pressure of 19.7 psi and temperature of -65°F.

Then

$$A = \frac{53.5 \qquad .144}{.61\sqrt{\dfrac{2(32.2) \quad (5)(144) \quad (3600)}{0.135}}} = .36 \text{ in.}^2$$

Inside diameter of a circular vent of the required area would be

$$\text{I.D.} = \sqrt{\frac{4(0.36)}{3.14}} = 0.675 \text{ in.}$$

The vent area necessary to prevent the tank pressure differential from exceeding 5 psi during climb at 25000 fpm., will now be found by first calculating the volume of air which must be discharged from the tank while climbing to 25000 ft., then calculating the vent area necessary for discharging this air at 5 psig.

$$\frac{P_1V_1}{T_1} = \frac{P_2V_2}{T_2}$$

where
P = absolute sea level pressure inside tank
V_1 = volume of air in tank at sea level
T_1 = absolute sea level temperature inside tank
P_2 = absolute pressure inside tank at 25,000 feet.
V_2 = volume at 25,000 feet of the same mass of air contained in tank at sea level.
T_2 = absolute temperature inside tank at 25,000 feet.

Since this is a 640 gallon tank,

$$V_1 = 640 \text{ gal.} \frac{1 \text{ ft.}^3}{7.48 \text{ gal.}} = 87 \text{ ft.}^3$$

Then

$$V_2 = \frac{T_2 \cdot P_1}{T_1 \cdot P_2} V_1 = \frac{P_1 V_1}{P_2}, \quad \text{assuming } T_2 = T_1$$

Since no pressure differential exists at takeoff,

$$P_1 = \text{atmospheric} + 5 \text{ psi} = 785 + 720 = 1505 \text{ psf}$$

Then

$$V_2 = \frac{2116}{1505} \cdot 87 = 122 \text{ ft.}^3$$

The rate of air discharge is therefore (122-87) = 35 ft.^{3}/min.

It is apparent that for this tank the specified fueling rate is the factor determining the vent area, since the rate of air discharge from the tank during fueling is greater than the rate of discharge during maximum rate of climb. Therefore, the minimum vent area should be .36 in.^{2}, and if a circular vent is used, its minimum inside diameter should be .676 in.

It should be noted that the effect of friction losses in the vent tubing was neglected in making these calculations. This effect would be quite small due to the relatively short length of tubing and would be more than compensated for in practice, since it is assumed that the vent diameter used would be the nominal size next above .676 in, or 3/4 in.
(7) WEIGHT: A breakdown of the estimated weight follows:

	Item	Wgt. - lbs.
1.	Outer shell	62
2.	Internal structure	125
3.	Cells	110
4.	Drain lines	16
5.	Float valve, pump, sump	15
	Total	328 lbs.

This weight is considerably more than the 143 lb. weight shown in Fig. 3 due to the fact that aluminum foil was used for the cell material instead of bladder material. However, this weight compares favorably with that of the present self-sealing tank, which is approximately 542 lbs.

VERTICAL TUBE TANK

The final design of the vertical tube tank is shown on Aeronautical Research Laboratory Drawing No. 6050 "Assembly-Tube Tank." A description of the design is as follows:

(1) OUTER SHELL: same as for cellular tank (see page 41) except for the heads. In the vertical tube tanks, the heads do not have internal integrally formed ribs for stiffening.

(2) INTERNAL STRUCTURE: The vertical tubes rest on the bottom of the tank in such a manner that no supporting framework is required except for the two center rows of tubes. These rows rest on an 18-gage (B & S) aluminum platform raised 2 in. from the tank bottom in the case of the center tube except those over the sump. The platform which supports the four tubes over the sump is raised 5 in. from the tank bottom in order to clear the pump motor housing. The space below these platforms is provided for lines connecting the tubes to the sump. Side loads on the tubes are carried by 20-gage (B & S) spacers which are spot welded together to form light-weight framework. To prevent collapse of the outer shell and to take the fore and aft loads from the tubes, four 18-gage (B & S) bulkheads with pressed lips are spot welded to the top half of the shell. The weights of the platforms, bulkheads and spacers are estimated at 9 pounds, 14 pounds, and 10 pounds respectively.

(3) TUBES: The tubes are 6-3/4 in. x 6-3/4 in. in cross section and made of corrugated aluminum foil. The tops are open except for a 1-in. lip folded inward to prevent splash--out. The bottom of the tubes are curved to fit the tank bottom except for the two center rows which are flat. The total number of tubes is 144, eight across the tank and 16 along the length of the tank. These tubes are assembled in a group of eight tubes to make a preconnected unit two tubes long and four tubes (one-half the tank width) wide. The average lengths of the tubes from the outside of the tank to the center are 14.5 in., 23.7 in., and 29 in. respectively. The length of the tubes over the sump is 26 in. Because the tubes set on the tank bottom (or on a platform for the two center rows) the drain is located in the side of the tube at its lowest point. A small channel formed by a step in the tube bottoms provides space for the drain lines. The estimated weight of the tubes is 65 pounds.

(4) DRAIN LINES: Each tube is fitted with a check valve to prevent a backflow from cells which might be elevated over the damaged cell. These check valves are shown in Aeronautical Research Laboratory Drawing #3425 "Valve-Check-Tube Tank."

The valves for adjacent tubes are built into one die-cast aluminum body which provides means of attaching a plastic tube to connect the unit of eight tubes. This plastic tube conveys the fuel to the main fuel line which is in the bottom of the tank and is also made of plastic. The main

fuel line runs the length of the tank and empties into a collector. The extimated weights of the plastic tubes and check valves are 14 pounds and 3 pounds respectively.

(5) FLOAT VALVE: Same as for cellular tank (See page 46.)

(6) VENT: Same as for cellular tank (see page 51)

(7) WEIGHT: A breakdown of the estimated weight is as follows:

(1)	Outer shell	62
(2)	Support platforms	9
(3)	Bulkheads	14
(4)	Spacers	10
(5)	Tubes	65
(6)	Drain lines	14
(7)	Check valves	3
(8)	Float valve, pump, sump	15
		192 pounds

SECTION IV

SUMMARY

The purpose of this study is to investigate various versions of compartmented fuel tanks under different battle-conditions in order to arrive at two alternate acceptable light-weight designs which will offer a reasonable protection against leakage. The drawings for the two designs are submitted separately from this report.

A compartmented fuel tank is one with internal divisions so arranged that the fuel is stored in separate compartments, each connected by a drain line to a sump in which is located a booster pump. The outer space, that is, all space between the compartments and the outer shell, is also connected to the sump. However, provisions are made for draining the outer space first, so that its fuel will have been used by the time combat conditions are attained. The reason for this is that any fuel leaking from the compartments due to damage will be collected in the outer shell and not lost externally.

Section I establishes the means for evaluating the leakage. A parameter is introduced, called the Percent Leakage. This is defined as the ratio between the amount of fuel lost to the amount in the tank before any damage. Methods for determining the Percent Leakage are developed by setting up a standard shot pattern. This pattern is subsequently used in evaluating all the designs under study.

Section II gives the results of evaluation for three style tanks: (1) The Cellular Tank, (2) The Horizontal Tube Tank, and (3) The Vertical Tube Tank. From the results of this evaluation, the Cellular Tank and the Vertical Tube Tank are selected for further study and design. (The latter title is shortened to Tube Tank for simplicity). The tank capacity chosen is 640 gallons.

Section III is devoted to a description of these two tanks. The Cellular Tank is made up of cubical compartments, each of which is connected to the sump by drain lines and check valves. (The check valve on a damaged cell prevents back flow from any cell which would be elevated with respect to the damaged cell.) The Tube Tank is composed of square vertical tubes, each of which is connected to the sump by drain lines and check valves.

Table 10 shows a comparison between the two designs.

TABLE 10

COMPARISON BETWEEN CELLULAR AND TUBE TANKS

Type	Wgt. lbs.	No. compartments	% LEAKAGE 100% Full	80% Full	60% Full	40% Full	20% Full
Cellular	328	780	29.0	12.0	8.0	3.0	0
Tube	192	144	24.3	15.4	12.5	0	0

Cell Dimension ---- 6 in. x 6 in.
Tube Dimension ---- 6-3/8 in. x 6-3/8 in.

It is to be noted that the Percent Leakages of the two tanks are approximately the same, yet the Cellular Tank contains 780 compartments, as against 144 for the Tube Tank. This is in disagreement with one of the design principles stated in Section I, "Design Criteria," namely, that the greater the number of compartments, the greater will be the leakage protection, or conversely, the lower the Percent Leakage. This design principle is true only if it is assumed that the fuel from each damaged compartment is lost. In the two designs studied above (the Cellular and Tube Tanks) this condition did not exist, since the outer shell or envelope could retain a large percentage of this fuel.

The comparison favors the Tube Tank slightly, due to the fact that one particular shot of the Standard Shot Pattern was so positioned that it fell on the edge of a row of cells in the Cellular Tank and damaged four rows instead of one. The same shot penetrated only one row in the Tube Tank, thus making the Percent Leakage a little lower than it should be. There is more external volume in the Cellular Tank than in the Tube Tank, which also has an influence on the Percent Leakages.

All in all, it appears that by using the proposed arrangement, in which fuel, leaking from a damaged compartment, can be retained by the outer shell, a tank with vertical tubes will have as great a leakage protection as one with cells. From a weight standpoint, the tube tank is superior, since the supporting structure of the Cellular Tank is not required. It is probable that filling and fuel metering problems are easier to meet with the Tube Tank.

Hence, the conclusions reached by this study are:

(1) The Vertical Tube Tank is lighter than the Cellular Tank for the same leakage protection.

(2) Compartmented Fuel Tanks can be made lighter and afford better leakage protection than present self-sealing tanks.

(3) Corrugated aluminum foil bonded together in layers can be made to form a suitable structural material for the compartments or tubes. The weight of this material would be of the order of 0.127 lbs/sq.ft.

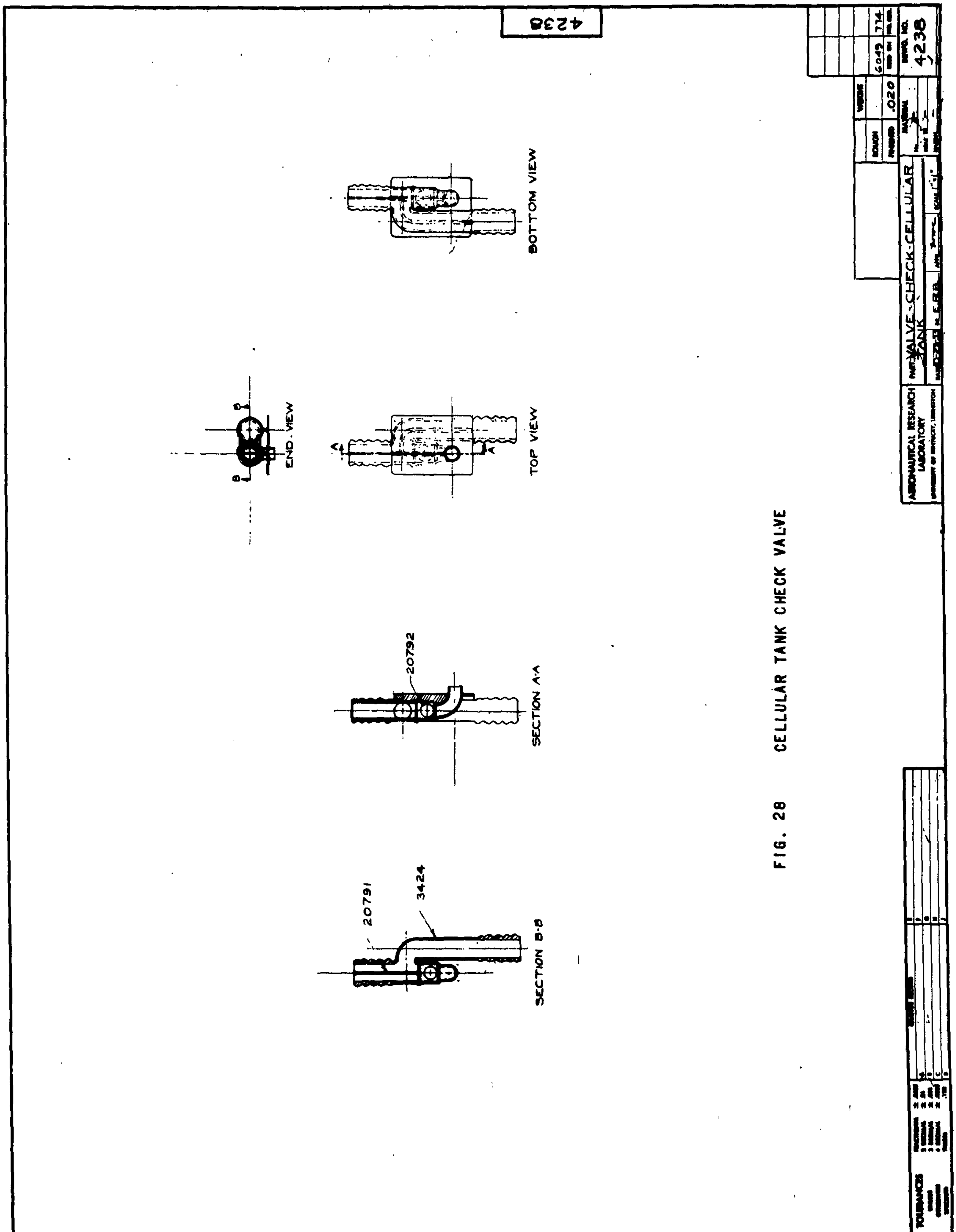


FIG. 28 CELLULAR TANK CHECK VALVE

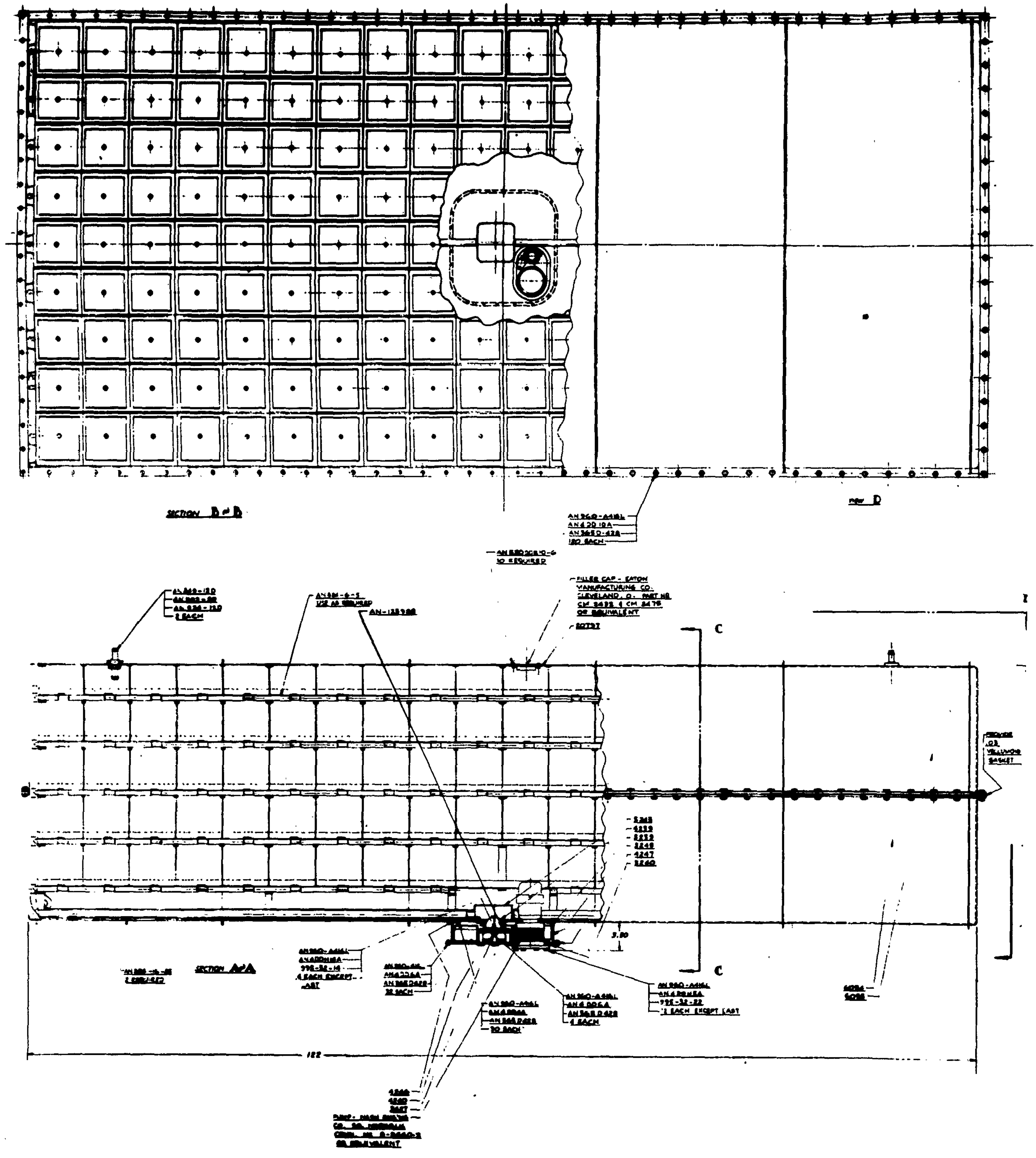


FIG. 29 CELLULAR TANK ASSEMBLY

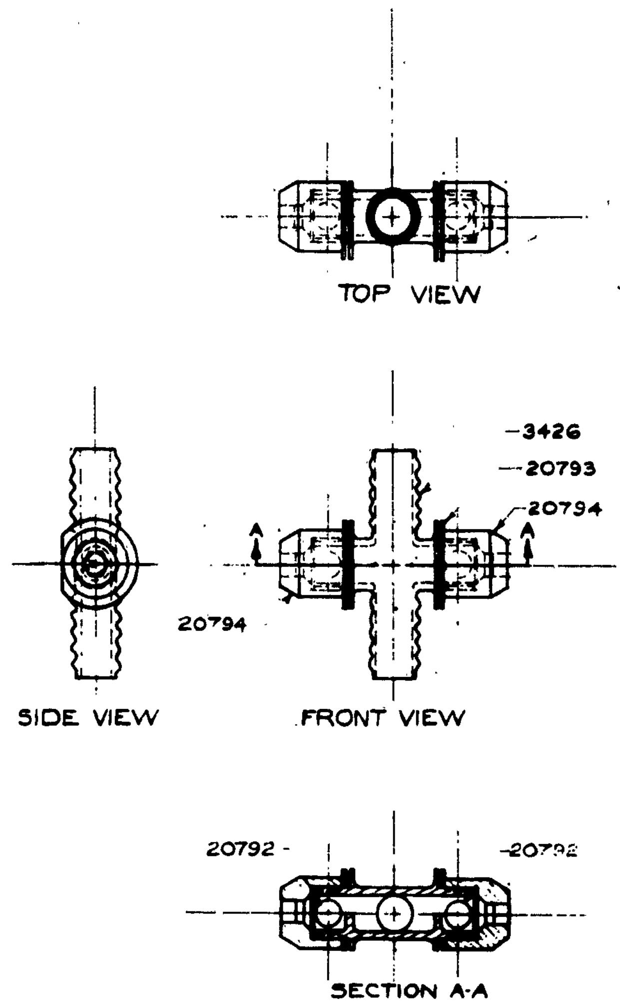


FIG. 30 TUBE TANK CHECK VALVE

AERONAUTICAL RESEARCH LABORATORY	PART VALVE-CHECK-TUBE TANK	DWG. NO. 3425
ROUGH 7	FINISHED .040	6050 72

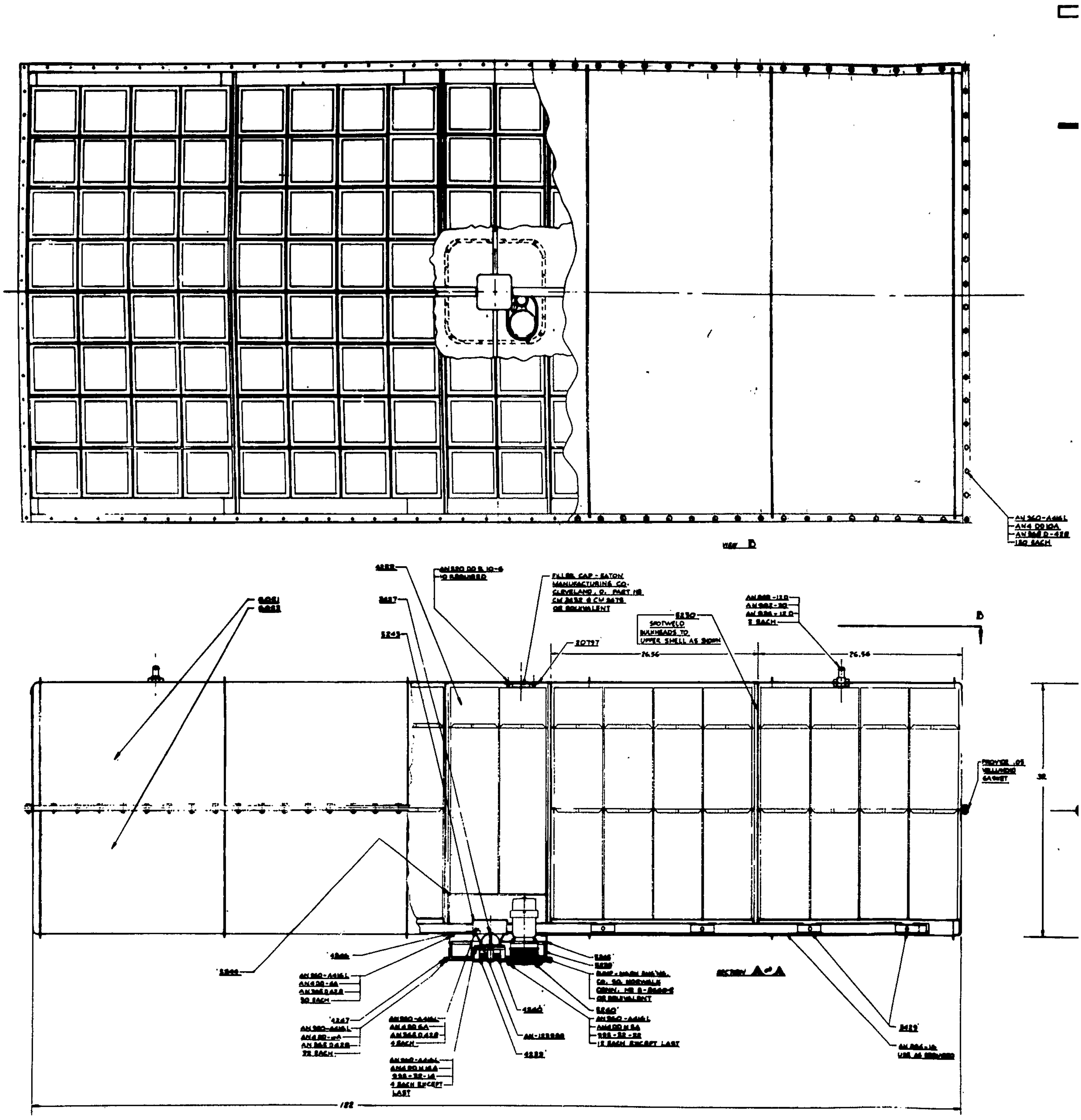


FIG. 31 TUBE TANK ASSEMBLY